“Performance and cost evaluation of Organic Rankine Cycle at different technologies”
(Master thesis)

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Abstract

ORC as a power generation method from medium and low thermal grade heat sources is currently a strong player in the market. Unprecedented energy demand, more environmental concerns and abundant low grade heat have motivated great actions for ORC usage. In this study, various fields which ORC can cost effectively be applied is evaluated and relevant cost reduction options are suggested.

By considering different applications, classification on produced power range and heat source temperature is implemented. Used medium, applied equipment and the cost of the system are investigated. Reasons on how the ORC can cost effectively fit into each system is presented. Influence of ORC application on cost, thermal and in some cases exergy efficiencies, system payback time, internal rate of return and net present value depending on rival technologies have been studied to support the idea. In section II with more details, case of rather low temperature grade ORC especially with solar based heat source is studied. A computer model is developed to compare thermal and exergy efficiencies, output power and turbine size factor as cost and performance indicators for different conditions.

Biomass and geothermal based ORC along with solar ORC with storage system can be considered as renewable resources in an acceptable cost to produce power. Waste heat recovery opportunities same as, waste heats in steel, cement, oil and gas, glass and vehicles industries and internal combustion engines are strongly recommended. Micro scale ORCs as modular units in home and office usages, remote areas power generation for about 2 billion people who do not have grid connected electricity in addition to solar desalination micro units and ORC in ocean thermal energy conversion are proven to be interesting applications of micro ORCs. ORC as bottoming cycle of recuperative or high pressure ratio gas turbines, micro gas turbines and fuel cells are also considered as other cost effective options. Moreover, it was shown that using ORC in combination with cooling and heating systems will improve the performance and decrease the cost of system. From computer model it was concluded that degree of superheats, evaporator and condenser pressures, fluid type and mass flow rates on ORC, heat sources and heat sinks sides, pumps efficiency, turbine types, etc, have significant effect on ORC performance and cost.
**Nomenclature:**

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ORC</td>
<td>Organic Rankine cycle</td>
</tr>
<tr>
<td>Tpp</td>
<td>Pinch point temperature</td>
</tr>
<tr>
<td>CHP</td>
<td>Combined heat and power</td>
</tr>
<tr>
<td>CCHP</td>
<td>Combined cooling, heat and power</td>
</tr>
<tr>
<td>HEF</td>
<td>Heat exchangers factor</td>
</tr>
<tr>
<td>SF</td>
<td>Turbine size factor</td>
</tr>
<tr>
<td>IC</td>
<td>Internal combustion</td>
</tr>
<tr>
<td>kW</td>
<td>Kilowatts</td>
</tr>
<tr>
<td>MW</td>
<td>Megawatts</td>
</tr>
<tr>
<td>ele</td>
<td>Electricity</td>
</tr>
<tr>
<td>Th</td>
<td>Thermal</td>
</tr>
<tr>
<td>Exeloss1</td>
<td>Exergy loss in evaporator</td>
</tr>
<tr>
<td>Exeloss2</td>
<td>Exergy loss in expander</td>
</tr>
<tr>
<td>Exeloss3</td>
<td>Exergy loss in condenser</td>
</tr>
<tr>
<td>H</td>
<td>Enthalpy</td>
</tr>
<tr>
<td>VCS</td>
<td>Vapor compression system</td>
</tr>
<tr>
<td>PGU</td>
<td>Power generation unit</td>
</tr>
<tr>
<td>SOFC</td>
<td>Solid oxide fuel cells</td>
</tr>
<tr>
<td>NPV</td>
<td>Net present value</td>
</tr>
<tr>
<td>IRR</td>
<td>Internal rate of return</td>
</tr>
<tr>
<td>PBT</td>
<td>Payback time</td>
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</table>

**Subscripts:**

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>State after pump</td>
</tr>
<tr>
<td>2</td>
<td>State after evaporator</td>
</tr>
<tr>
<td>3</td>
<td>State after turbine</td>
</tr>
<tr>
<td>4</td>
<td>State after condenser</td>
</tr>
<tr>
<td>eff0</td>
<td>Optical efficiency of collector</td>
</tr>
<tr>
<td>a,b</td>
<td>collector constants</td>
</tr>
<tr>
<td>T0</td>
<td>Ambient temperature</td>
</tr>
<tr>
<td>Tint</td>
<td>Heat transfer fluid temperature</td>
</tr>
<tr>
<td>( \delta T' )</td>
<td>Degree of superheat in evaporator</td>
</tr>
<tr>
<td>Tpp</td>
<td>Pinch point temperature</td>
</tr>
<tr>
<td>Tpp1</td>
<td>First Evaporator pinch point temperature</td>
</tr>
<tr>
<td>Tpp2</td>
<td>Second Evaporator pinch point temperature</td>
</tr>
<tr>
<td>Tppc</td>
<td>Condenser pinch point temperature</td>
</tr>
<tr>
<td>Twater3</td>
<td>Intermediate fluid exhaust temperature from pre-heater</td>
</tr>
<tr>
<td>PEMFC</td>
<td>Proton Exchange Membrane Fuel Cells</td>
</tr>
<tr>
<td>MOFC</td>
<td>Molten oxide fuel cells</td>
</tr>
</tbody>
</table>
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1: Applications

1- Introduction

Nowadays due to 40% predicted increase in energy consumption of the world, more environmental concerns and less dependency on fossil fuels, demand for sophisticated power supply options is greatly increased. Environmental concerns will be in form of global warming, acid rains, air, water and soil pollution, ozone depletion, forest devastation and radioactive substances emissions.

Utilizing waste heats along with attempts to derive energy out of renewable resources as low grade thermal heat sources have motivated the use of ORC.

ORC or Organic Rankine cycle basically resembles the steam cycle according to working principles. In ORC, Water is replaced with a high molecular mass fluid with lower degree of boiling temperature in comparison with water. Fluid characteristics make ORC favorable for applications of low temperature heat recovery (normally less than 400°C).

In Figure 1, a T-S diagram of ORC is depicted. As normal ORC, pre-heater (2-3) which preheats the working fluid, evaporator (3-4) which changes the phase of working fluid from liquid to vapor by gaining heat from the heat source, super heater (4-5) which will superheat the working medium by adding more heat from heat source, turbine (5-6) which expands the working fluid and extract power from it, de-superheater (6-7) which will form saturated vapor fluid at condenser inlet from superheated fluid after turbine, condenser(7-1) that removes the heat from fluid and make it liquid and pump (1-2) that increases the liquid pressure are main components. Entire process can be repeated to produce continuous power.

![Figure 1: Normal ORC T-S diagram](http://en.wikipedia.org/wiki/File:Ideal_and_real_organic_rankine_cycle.jpg)

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These and additional components like regenerator will be more studied in chapter II-1, “More detailed case studies – Cost reduction and performance improvement of ORC” and also in appendix, section A-2.

ORC has several advantages over steam cycle. It is known that working fluids in ORC has higher molecular weight than water. This will increase mass flow rate of fluid for the same sizes of turbine. More mass flow rates will give better turbine efficiencies and less turbine losses (Drescher, D. Bruggemann, 2007). Moreover, turbine efficiency which is about 85% is kept efficient in part load applications and the system can be started faster. Most importantly, boiling point of ORC fluids are less than water; hence, they can be applied in lower temperatures.² (For more information about mass flow rate in turbine and losses please see the appendix A-2, turbine losses).

There are some other alternatives for power generation from low temperature heat sources. Kalina cycle and transcritical CO₂ power cycles are two important examples.

Due to gliding temperature for Kalina cycle working medium in evaporation and decreasing temperature during condensation, better heat transfer from heat source and to the heat sink is expected. This causes better efficiency for Kalina cycle. However, to achieve this, higher maximum pressure shall be maintained in the Kalina cycle in comparison with ORC and this makes the cycle to be more expensive. More components same as absorber and separator is needed, turbine needs to be multistage or have a high rotational speed and since the medium is a mixture and normally is from water and Ammonia, it is deemed to be corrosive. All in all, these will result in bulkier system, more components and higher electricity price from Kalina cycle in comparison with ORC.³

Transcritical power cycles can also be used to generate electricity from low temperature heat sources same as vehicles exhaust, geothermal and solar energy resources. Phase change, same as Kalina cycle, occurs in non-constant temperature which will enhance the heat transfer. One of important working mediums is CO₂ with 4-10 times more slope in vapor pressure diagram in comparison with many fluids which will give proper heat transfer criteria for the cycle.

Transcritical power cycle produces more power than ORC and gains better efficiencies. One important obstacle on the way of its improvement was that the medium has properties in between liquid and the gas. This delayed the development of the cycle due to lack of appropriate expander which works in variable phase situation. Due to very recent achievements in variable phase turbines this technology is available today. Arrangement of the turbine is same as normal impulse axial turbines; however, special shape of the blades will pass the small liquid droplets with the gas and will minimize the erosion. In Figure 2 the variable phase turbine blade and a normal turbine blade have been depicted.

In comparison of a real geothermal project using R134a as a variable phase cycle and an ORC based geothermal plant, it was concluded that the overall cost including the power cycle can be brought down from 4000$/kWe in ORC based technology to 3000$/kWe in variable phase power cycle.³

Nano-structured metal-organic heat carriers can also be used to enhance the ORC overall performance and reduce the ORC cost. US Department of Energy’s Pacific Northwest National Laboratory (PNNL) is one of players in research and developments on the issue. A new project on low temperature geothermal based power generation is under investigation by the aforementioned institute.⁵

Nano-structured metal-organic heat carriers are Biphasic fluids with rapid expansion and contraction capabilities. When they get heat, the thermal-cycling of the biphasic fluid will run a turbine to produce electricity. Due to their features and better heat delivery from heat source, they can produce more power in ORC unit and they can increase the power generation capacity to near that of a conventional steam cycle. By applying them cost per produced unit of electricity can be reduced. Their development is under study and it is hoped that they will be more known and applied in near future (Phil Welch, Patrick Boyle, 2009).⁴

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1-1 Purposes

Demand for sophisticated and affordable power generation systems, security of electricity supply in each country and environmental concerns encourage the investigation for cost effective ORC. It is an environmental friendly option to meet this important need. Rural electricity provision for nearly 2 billion people and better waste heat recovery of different industries in addition to Renewable based power generation can benefit from this.

Moreover, a comprehensive study on different applications of cycle in different technologies cannot be found and preliminary study shows that ORC applications have been overlooked in many areas. These all give great impetus for this study. As a result, this master thesis is aimed to be a study on economically feasible applications of Organic Rankine cycle and solutions to improve its performance and cost.

It is targeted that a market analysis be performed to identify cost effective applications of cycle in different technologies. Classification on produced power range and heat source quality and temperature is set to be implemented to recognize the technology, used working fluids and applied equipment.

Each component and process of the cycle in introduced technology will be studied and suggestions will be made to reduce the overall cost, improve the efficiency and power output of cycle. Recommendations for enhancement of solar based ORC system should be given in rather detailed manner. Introduced improvement options are better to be expandable for many other applications to promote the system performance and reduce the costs.

1-2 Overview on Methodology

By searching internet commercial websites, digital libraries, brochures from manufacturer and via contacting related companies, necessary data is collected. It includes costs, configurations, power and temperature ranges, and special considerations regarding each ORC technology.

Then the gathered data has been sorted and classified. During the evaluation, life cycle cost of ORC, quality and quantity of produced energy and ORC properties as well as its initial cost has been taken into account. Then the components of ORC have been well recognized and some recommendations have been proposed for cost reduction opportunities. In each application, other technologies which can be used alternatively have been introduced and compared with ORC. This causes better understanding of whole system and to identify most proper technology in each application. In some cases the comparison shows the advantages of ORC more clearly.

To reach aforementioned goals, equipment of ORC shall be selected properly. The working fluid, ORC speed of turbo machineries, degree of superheat, working minimum and maximum temperatures and pressures are checked in data recourses for some applications.
A computer model for thermal and exergy efficiencies, heat transfer factor and turbine size is built. The outputs give information on cycle performance and cost as mostly some non dimensional numbers. Then, effect of selecting components with different characteristics and ORC features has been studied on them.

1-3 Background

It is believed that idea of ORC as power generation system from low heat sources was first presented in 1961 by Israeli solar engineers Harry Zvi Tabor and Lucien Bronicki. \(^6\)

In Figure 3 it is shown that current market share of Biomass ORC is a dominant share and Geothermal which used to be the most significant part is in second position currently. Waste heat recovery in different industries which is in third place can be applied to numerous industries and solar ORC still have a huge potential to be flourished. This technology, due to lack of awareness, currently is only 1% of ORC market. Biomass based application is so developed since it is the only proven technology for decentralized applications to gain powers up to 1MWel from solid fuels like biomass. \(^10\)

From 1980s which ORC has been available in the market, more than 200 projects with total power of about 2000 megawatts of electricity have been installed. This is while that rate of ORC usage has been unprecedentedly soared in recent years. This shows a promising future for ORC market in near future.

![Figure 3: ORC market share for different heat sources](http://www.labothap.ulg.ac.be/cmsms/uploads/File/ECEMEI_PaperULg_SQVL090407.pdf)

Source: Fifth European conference, Economics and management of energy in industries. \(^7\)

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1-4 Initial classification

After evaluation of gathered information, preliminary calcification of ORC applications is summarized as here bellow:

- Biomass based ORC
- Micro scale ORC which includes small CHP ORC, Solar desalination unit, ORC as micro gas turbine bottoming cycle and Ocean thermal energy conversion technology for ORC applications
- Geothermal ORC
- ORC for waste heat recovery of some industries, e.g.: Steel, cement and ceramic industries or internal combustion engines or vehicles waste heat recovery
- ORC as bottoming cycle of Gas turbine
- Fuel cell based ORC
- Solar ORC –Larger solar power plants using ORC with possibility of storage systems
- ORC in nuclear power plants
- ORC for cooling

In Table 1 several main ORC producers, power range, temperature range and brief used technology is mentioned.
Table 1: Classification of ORC application based on power, heat source temperature and producers

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Applications</th>
<th>Power range</th>
<th>Heat source temperature</th>
<th>Technology</th>
</tr>
</thead>
<tbody>
<tr>
<td>ORMAT, US</td>
<td>Geothermal, WHR, solar</td>
<td>200 KWe – 72 MWe</td>
<td>150° - 300°C</td>
<td>Fluid : n-pentane</td>
</tr>
<tr>
<td>Turboden, Italy</td>
<td>CHP, geothermal</td>
<td>200 KWe – 2 MWe</td>
<td>100 - 300°C</td>
<td>Fluids : OMTS, Solkatherm Axial turbines</td>
</tr>
<tr>
<td>Adoratec, Germany</td>
<td>CHP</td>
<td>315 – 1600 KWe</td>
<td>300°C</td>
<td>Fluid: OMTS</td>
</tr>
<tr>
<td>GMK, Germany</td>
<td>WHR, Geothermal, CHP</td>
<td>50 KWe – 2 MWe</td>
<td>120° - 350°C</td>
<td>3000 rpm Multi-stage axial turbines (KKK) Fluid: GL160 (GMK patented)</td>
</tr>
<tr>
<td>Koehler-Ziegler, Germany</td>
<td>CHP</td>
<td>70 – 200 KWe</td>
<td>150 – 270°C</td>
<td>Fluid: Hydrocarbons Screw expander</td>
</tr>
<tr>
<td>UTC, US</td>
<td>WHR, geothermal</td>
<td>280 KWe</td>
<td>&gt;93°C</td>
<td></td>
</tr>
<tr>
<td>Cryostar</td>
<td>WHR, Geothermal</td>
<td>n/a</td>
<td>100 – 400 °C</td>
<td>Radial inflow turbine Fluids: R245fa, R134a</td>
</tr>
<tr>
<td>Freepower, UK</td>
<td>WHR</td>
<td>6 KWe - 120 KW</td>
<td>180 - 225 °C</td>
<td>Turbo-expander</td>
</tr>
<tr>
<td>Tri-o-gen, Netherlands</td>
<td>WHR</td>
<td>160 kWe</td>
<td>&gt;350°C</td>
<td></td>
</tr>
<tr>
<td>Electratherm, US</td>
<td>WHR</td>
<td>50 KWe</td>
<td>&gt;93°C</td>
<td>Twin screw expander Fluid: R134a Radial Turboexpander</td>
</tr>
<tr>
<td>Infinity Turbine</td>
<td>WHR</td>
<td>250 KWe</td>
<td>&gt;80°C</td>
<td></td>
</tr>
</tbody>
</table>

Infinity turbine produces 250 KWe ORC and recently 10 KWe ORC boxes from this company are available in the market. Free power in UK is another producer of small ORC units. Opcon in Sweden can also be added to this endless list. ORMAT and Turboden are some other important manufacturers.

In next chapters applications of ORC will be studied.

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2- Biomass related applications of ORC, rather higher grade heat applications

Due to high availability of Biomass as solid fuel for power generation, this option has been increasingly under development. Rather proper part and full load efficiency at small scale biomass based power generation plants attracts attention for ORC usage. Moreover, ORC cycle acts more economically in smaller powers since less safety measures are applied at lower powers and temperatures. As a result, Biomass based ORC has a large market share of the technology beside Geothermal and waste heat recovery applications.

Biomass based ORC differs in some aspects with usual known ORCs and this requires accurate and elaborative study on the subject.

Boilers in biomass applications are heated up to 450°C and 60 – 70 bars which require expensive materials in design of equipment. When ORC is used in the system, biomass boilers are heated up to 300°C which are less costly than normal biomass boilers. However, this is rather high temperature in ORCs. This will give a great opportunity of using the system as combined heat and power cycle which will be discussed later.

Cooling side of condenser in biomass applications can be used in district heating systems as heat supply. This makes another difference with normally practiced ORC applications since the temperature of condenser may reach up to 100°C.

Carnot efficiency for typical biomass based ORC with heat source temperature of about 300°C and heat sink temperature of 100°C is about 35% which means that this application has higher potential for thermal efficiencies in comparison with normal ORCs with Carnot efficiencies of usually about less than 25%.

One important restriction is the maximum allowable working temperatures of about 400°C. In these high temperatures working fluids are prone for dissociation and instability due to chemical reactions in working fluids which occur in high temperatures. Furthermore, the maximum pressure in Biomass ORC is normally kept bellow 20 bars to avoid expensive safety measures and materials. If large amount of heat is available, superheating will be the solution to keep the higher pressure of system bellow 20 bars.

In ORC applications better efficiencies are when less superheat is done and reasonable higher evaporation temperature and pressure are selected. This will be well discussed in (SectionII: chapter 1). To keep the superheat in an acceptable range, the system should be optimized for maximum pressure (SectionII: Chapter 1), of section and for biomass based operations, around 20 bars is normally suggested.

For biomass based ORC applications, using Alkylbenzenes family and specially Butylbenzene as working fluid are assumed to give best efficiency.
The aforementioned issues related to Biomass based ORC usage is summarized in table 2 and some considerations have been added to give important suggestions on cost reduction options.\(^9\)

<table>
<thead>
<tr>
<th>Items to be implemented</th>
<th>Keep the maximum working pressure bellow 20 bars</th>
<th>Maximum working temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reasons for cost reduction</td>
<td>Less safety measures will be needed, less material cost</td>
<td>Increases efficiencies</td>
</tr>
<tr>
<td>Considerations</td>
<td>Excessive superheat shall be prohibited to keep evaporator pressure high enough* However, in case of high available heat, to keep the evaporator max. pressure bellow 20 bars superheat will be done**</td>
<td>Optimization should be performed. Excessive increase in not interesting.*</td>
</tr>
</tbody>
</table>

*For more details see chapter II-1.

**Available heat source can be used in both evaporator and super heater. If more fraction of heat is used in evaporator, higher temperature and pressure in evaporator will be achieved. Since evaporator pressure above 20 bars will raise the cost of material and safety measures, if this pressure is reached in evaporator, more fraction of input heat will be used in super heater. Otherwise, it is interesting to limit the superheat. This can be done by designing the evaporator and super heater and pressure after the pump in a well manner. For super heating effect and adjusting pump pressure see chapters II-1 and II-6.

### 2-1 Comparison of ORC in biomass CHP plant with gasification and IC engines

In biomass gasification, biomass will be converted to Hydrogen and Carbon Monoxide (Syngas) which is a suitable fuel. Gasification is an efficient and rather low cost energy conversion method to harness biomass energy and produce electricity.(For more information see APPENDIX: A-2).

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Another option to extract biomass energy is biomass combustion. As depicted in Figure 4, the biomass is combusted and a medium (Here thermal oil) delivers heat to vapor generator of ORC unit. Then power will be produced in ORC unit. Waste heat of ORC condenser can be used in district heating system. This heat can also be used in an absorption chiller (tri-generation) to produce cooling as well as heat and power, depending on demands. Even produced power from ORC can be used in vapor compression chiller. Adding district heating and absorption chiller will increase the total thermal efficiency of whole plant. In Figure 4, ORC fluid after expander will be used in regenerator. In this device (regenerator), remained amount of heat in ORC fluid after turbine will be used to pre-heat the fluid before entering the ORC evaporator (vapor generator).

In a typical plant with 30 MWh/year heat production and net generated electrical power of 4.6 MWh/year, total efficiency of more than 80% is achieved and using 38.240 MWh/year of natural gas is avoided that results in the prevention of carbon-dioxide emissions to about 10000 tons per year (A.Rentizelas, S. Karellas, E.Kakaras, I.Tatsiopoulos, 2009).

The Figure 4 shows how the ORC system will be fit in the combined heat and power system based on biomass combustion.

![Figure 4: Combined heat and power plant using ORC in combination with Biomass combustion system](image)

Source: Energy Conversion and Management 2009

Combined heat and power plant using ORC in combination with Biomass combustion system can be compared with combination of biomass gasification and internal combustion engines. In Figure 5, gasification system with IC engines is depicted. In the latter system (gasification based system), biomass is converted to Syngas (CO and

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H₂) in gasifier, and then the produced gas (fuel) is cleaned and is burned in an IC engine. Waste heat of the system can be used for district heating. Both systems can produce heat and power; however they differ in some aspects as here bellow. (Definition of gasification is mentioned in Appendix, section A-2).

First, electricity to heat ratio in gasification and IC engines is more than double in comparison with CHP plant using ORC in combination with Biomass combustion. Gasification based system needs about 30 - 40% more biomass to achieve rather same heat production.

Secondly, since gasification deals with higher temperatures, more losses occur in heat transfer system, hence, more nominal power is expected to compensate losses and give the same heat output. To clarify this, forward heating pipe in gasification type is about 120 °C and in ORC type is 90°C.

All above mentioned issues will give a little more total efficiency for combined heat and power cycles using ORC rather than gasification (A.Rentizelas, S. Karellas, E.Kakaras, I.Tatsiopoulos, 2009), however, better electrical efficiency in gasification based case will make the industry owners to select the technology based upon their electricity or heat demand.⁹
Considering electricity and heat prices as an example in Greece\textsuperscript{10}, due to more electricity generation in gasification based system, this technology gives twice net present value in comparison with ORC based CHP plant (A.Rentizelas, S. Karellas, E.Kakaras, I.Tatsiopoulos, 2009); however, much more initial investment is needed for gasification based system which means not so lower payback time and not so higher internal rate of return for the technology. Definition of terms “net present value”, “internal rate of return” and “payback time” is mentioned in section A-2. \textsuperscript{10}

ORC in combined heat and power system has about 20\% less internal rate of return and about 25\% more payback time in comparison with gasification based system. Gasification has about two times the net present value since the effect of revenue from electricity generation is more important in this economical factor. The Figure 6 shows the initial cost break down for two technologies. \textsuperscript{9}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure6.png}
\caption{Comparison of ORC and gasification based combined heat and power plant based on initial cost}
\label{fig:figure6}
\end{figure}

\textit{Figure 6: Comparison of ORC and gasification based combined heat and power plant based on initial cost}
\textit{Source: Energy Conversion and Management 2009}\textsuperscript{9}

The Figure 7 Shows the return of investment for produced heat, cooling and electricity in both technologies.
District heating system in case of ORC needs higher pipe diameters since the temperature is lower. This increase is about 5-6% in cost. However, more necessary biomass preparation will cost about 40% more in gasification case. Furthermore, ORC is much more known system and is available in packages which make it easier and safer to work with. This system has half cost for operation and maintenance and has less operational risk in comparison with Gasification based technology. ORC only needs a weekly operator inspection and just better to be checked twice in a year by producer.

To conclude from economical comparison, ORC in combined heat and power technology has a little less internal rate of return and just small amount of more payback time in comparison with gasification based technology. As mentioned, due to more electricity production, gasification based system will return more revenue, has higher net present value and is more interesting when long time investment, more electricity and more revenue is considered over 20 years of its lifetime.
However, heat to electricity price ratio will help in selecting the right technology. Closer prices for heat and electricity, more demand for heat, interests for lower risk, more safety, less installation time and low available initial budget might give enough impetus for ORC application and to disregard high income and more electricity generation from gasification system.  

### 2-2 Biomass digestion plant waste heat recovery

Currently extracting biogas energy in biomass digestion process is not so economically active and efficient method. The most important reason is huge amount of waste heat which cannot be used in district heating system since the plants are normally in remote areas and it is not practical to transfer heat economically for rather far distances.

Using ORC and producing electricity from biomass digestion plant waste heat can improve the efficiency of the plant, produce base load electricity and help in more clean power generation.

In Figure 7 a schematic of an already installed system is depicted.
According to Figure 8, extracted biogas from Biomass (2) is combusted in engines (1). Compressed air from turbo generator (3) and cooling water heat is utilized to heat the digester. Oil will be used to extract heat from flue gas of combustion (9) which will give heat to ORC fluid (10-12). The ORC fluid can be heated or even super heated normally in a single heat exchanger which is well known as evaporator (11-12).

An implement cost evaluation on added ORC system shows that for a typical 500KW biomass digestion plant, additional 35KW electricity can be produces by an extra investment of 4800$/kW. Assuming more than 7500 hours of full operation, 15 years of life for system and interest rate of about 5%, it will yield 0.074 $/kWh electricity which is quiet lower than normal biomass based electricity price of about 0.17 $/kWh.\textsuperscript{11}

3- Micro scale application of ORC down to 1KW

3-1 Micro scale CHP, mainly solar and biomass based, ORCs

In many places of the world, same as Africa and south Asia, great number of people do not have access to grid connected electricity. The population nearly reaches 2 billion people.

Heat from Solar, biomass and geothermal energy or waste heats in small scales can help in providing clean and cost effective electricity in remote areas. Using ORC in this concept will be a turning point.

Moreover, goals to reduce green house gas emission till 2020, has motivated large investment in new cleaner sources of energies. Buildings as one of main energy consumers can also benefit from ORC. Electricity from solar energy is one solution. Photovoltaic systems which have the greatest share of current market are rather costly to be used. Besides, more flexibility and availability of power source persuade other solar based technologies:

One novel idea is to use gas and solar hybrid combined heat and power cycles in very small scales for domestic and even remote houses or in offices. This system main energy source is solar energy which is abundant in many places. Since ORC unit input is heat, when solar energy is not available a gas burner or other heat source can be added to increase the availability of system. It is a cheaper solution than PV based electricity generation. Cost and performance of solar based ORC will be compared with solar P-V in chapter 9 and more details are given in SectionII, 2.

To clarify the micro ORC advantages it is worthwhile to mention that square root of optimum power for ORC is proportional to inverse of ORC speed which is turbine pump rotational speed. Weight to power ratio will be more in case of higher powers since the ORC speed will be less. In other word, it is beneficial to optimize number of installed ORCs units rather than using single unit to reach interested power while the cost of additional unit and additional produced power is considered.

All in all, Ability to fulfill undeveloped and remote areas needs of heat and electricity, incorporating waste heat utilization in overall system and less losses and costs for electricity transmission lines will motivate the use of micro scale ORCs which can be driven by heat from solar energy, small amount of gas, biomass, waste heat, etc. In Figure 9, one set of designed micro combined heat and power is depicted. ORC is the core of system. Moreover, In Figure 24, another CHP plant which uses micro ORC as supplementary power generation system is shown, however on that system boiler is not using solar energy directly.
Illustrated system in Figure 8 can be run by a heat source down to 25kW. When the energy from sun suffices, the system uses only solar energy. The alternative heat source which might be a gas burner increases the availability of the system while solar energy is not sufficient. It can replace the solar energy and produce up to 100% heat input (25kW).

Evacuated glass tube solar collector is connected to compact brazed heat exchanger which is used as evaporator of ORC unit. A heat pump is added to solar collector to use its waste heat which is not used in ORC evaporator. The pump of this unit needs to be accurate; as a result a double diaphragm pump is chosen. Turbine speed is 60000 rpm here. The ORC working fluid in this power range and low temperatures of about 90°C is chosen to be HFE-301. This fluid has low boiling temperature in 3-4 bars as 75°C, is not flammable as

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other equivalent fluids and is relatively dry after expansion. This fluid is not listed as volatile organic compounds which influence climate significantly and has no ozone depletion potential (gpo.gov, 2002). ORC evaporator will generate vapor of 3-4 bars and then run the micro turbine rather efficiently.13

Normally 10% of input heat (typically 2.5kW) can be recovered in condenser and be used for heating purposes same as space heating. This gives total cycle efficiency of 17% and electrical efficiency of system is about 7.5%. More details about this system is given in SectionII, 2.11

3-2 Cost of the system
Results from an economical analysis (A. Schuster, S. Karellas, E. Kakaras, H. Spliethoff, 2009) on rather the same micro CHP unit with 5kW electricity production and efficiency of 8% are shown in economical performance curve in Figure 10. The here bellow graph shows electricity production price based on heat price and operational hours. In this Figure, vertical line depicts operational hours, horizontal line shows heat price. (Heat price/revenue shows the price of heat, since this system consumes heat and some amount of its waste heat (typically 10%) is used for heating purposes, this number affects its total revenue). Price of produced power is shown in Ct (1Ct means 0.01 €/KWh). Using the system for about 7500 hours will yield about 0.105€/KWh for electricity production while typical heat price is assumed to be 0.09€/KWh.12

\[\text{Figure 10: Electricity price for a 5kW micro ORC system (1Ct means 0.01 €/KWh)}\]
\[\text{Source: Applied Thermal Engineering.}^{14}\]

Micro ORC CHP unit applications in remote areas which uses solar or biomass energy as heat source will decrease the electricity price down to about 0.14 $/KWh that is quiet lower than diesel based electricity price of 0.3$/KWh. They can be a solution to heat and electricity demands and will significantly help in decreasing the CO₂ emission. This is another advantage of system which decreases the tax of system and can have revenues for clean electricity generation with low emission (Infinity turbine 2010).

One other great improvement in micro ORC technology is 10KW ORC power boxes. They are compact, safely operational systems and commercially available technology. They can be joined to many heat sources same as waste heat recovery in industries, solar collector or small biomass boilers. A system with its specifications and cost is showed in Table 3 (Infinity turbine, 2010).

<table>
<thead>
<tr>
<th>Technology</th>
<th>Produced power range</th>
<th>Heat source temperature range</th>
<th>Turbines</th>
<th>Heat Exchangers</th>
<th>Working fluid</th>
<th>Size</th>
<th>Company or institute working on</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Micro ORC</td>
<td>10KW</td>
<td>Around 100°C</td>
<td>Lysholm Turbine – 60% of cost</td>
<td>Compact brazed heat exchangers</td>
<td>R134a, R245fa, R22, Other refrigerants</td>
<td>0.6<em>1.5</em>1.5 (m³)</td>
<td>Infinity turbine</td>
<td>25000$ - 2500$/KW</td>
</tr>
</tbody>
</table>

Table 3: A 10kW ORC box information

Source: Infinity turbine. ¹⁵

3-3 Small scale ORC for solar desalination units

Great progress on membrane technology has made reverse osmosis as pioneer option in desalination systems. In this technology, a selective membrane removes large molecules from the pressurized solution. To apply pressure on solution, a pump is used. This pumps requires power to be run and this can be supplied by an ORC unit.\(^\text{16}\)

One good incentive to use ORC as driver of desalination unit is that high pressure pump of saline water to membrane unit can be coupled with ORC turbine directly (EX is coupled to HPP in Figure 11) and this reduces the losses in conversion systems which convert electricity to rotational speed and vice versa. To clarify this, in PV based desalination unit electricity is produced and this will run the pump, however, in system of Figure 11 since turbine is coupled with turbine directly, generator is not used and rotational speed is inserted to pump directly and this is a more efficient way.

So far using parabolic trough system and screw expander as driver has been best economical option even more appropriate than PV driven systems (Can be 40-60% less expensive than P-V based system).\(^\text{55}\)

To have a solar - ORC desalination unit as in Figure 11, a collector system, ORC and desalination unit shall be connected. The heat transfer medium in collector system is pumped water which will deliver the heat from collector (ETC) to ORC evaporator (EV). The turbine in ORC subsystem (EX) will drive a high pressure pump (HPP) from desalination subsystem and this pump will provide water to membrane (RO). A hydraulic turbine (HT) will use remained pressure of fluid coming from the membrane to run a small pump (P3).\(^\text{14}\)

![Diagram of Small scale solar – ORC desalination unit](source:image)

**Figure 11: Small scale solar – ORC desalination unit**

*Source: Applied Energy\(^\text{16}\)*

Performing an exergy analysis and defining a term called thermodynamic performance will help in analyzing different configuration and fluids for the system. Thermodynamic performance will be defined as exergy of outgoing flow to energy input of each sub-system. This will show importance of evaporator and proves that

biggest exergy loss occurs in expander due to thermal and radial leakage losses. Besides, it is shown that (B.F. Tchanche, Gr. Lambrinos, A. Frangoudakis, G. Papadakis, 2010) regenerator (device which transfers heat from vapor fluid after expander to liquid fluid going to evaporator) especially in isentropic fluids like R134a will not add to better performance of system; because, it is seen that in some performed experiment the fluid temperature after regenerator will be less than fluid temperature in pump outlet.

3-4 ORC as bottoming cycle of micro gas turbines

Environmental and energy demand concerns especially in non interruptible applications same as in hospitals or remote areas has caused an unprecedented interest in micro turbines application for power generation. Micro gas turbines as rival for internal combustion engines are one potential option. Lower efficiencies of about 30% have motivated more research on the issue. One genius solution is combining the micro gas turbine with ORC to reach higher efficiencies of about 40%. This is even true for small power generations of about 100 kW for micro gas turbines and about 35 kW for ORC. In more general view, micro gas turbines gas power can be up to 150kW. ORC power will increase accordingly since the waste heat from micro gas turbine will increase with its power.

In Figure 12 it is shown that exhaust gas from gas turbine which has temperature of about 250-300°C is combined with an ORC unit to produce extra power. The heat transfer from gas turbine to ORC evaporator occurs at heat recovery vapor generator (Red area).

![Figure 12: ORC as bottoming cycle of a micro gas turbine](source: Alureiter; Wikipedia)

For this technology choosing of working fluid and best turbine are the key points. Except than some physical properties of fluid, normally the fluid selection is done based on turbine exhaust gas temperature. In
evaporator pressure, the more the boiling temperature of fluid is closer to exhaust gas temperature the better power output will be the result (Costante Invernizzi, Paolo Iora, Paolo Silva, 2007).

With respect to turbine efficiency, normally two factors are considered. Ratio between volumetric flow rate of outgoing and incoming fluid in addition to size factor are key parameters in choosing the right turbine. This will be discussed in chapter I-1. Some candidate working fluids for this application can be found in Costante Invernizzi, Paolo Iora, Paolo Silva, (2007). For gas turbine exhaust temperatures of 300°C, esa-methyl-disiloxane is suggested by authors.

Based on normal ORC cost of about 3500$/KW in 50 – 100 KW power range and micro gas turbine cost of about 1500 $/KW, total system cost will be about 2100$/KW. By this investment efficiency of the system will be increased from 30% to about 40.

3-5 Ocean thermal energy conversion technology

Ocean thermal energy can be utilized to drive an ORC. Warm see water can be used in evaporator while colder water in deeper parts of sea can be used in condenser. Small scale ORCs with possibility for low thermal heat source usages are suitable for this option.

Temperature differences in evaporator and condenser are normally low and in order of 20°C. Working fluids can be refrigerants and benzene series fluids. This technology is in demonstration phase and is not highly commercially attractive yet (S.K. Wang, T.C. Hung, 2010).

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4- Geothermal applications

4-1 Low enthalpy geothermal applications:

Geothermal energy as stored energy in earth is assumed to be a renewable source to produce clean electricity especially in remote and prone areas. Normally the geothermal wells have temperatures of between 50 – 350 °C.

To gain higher temperatures in earth, deeper wells are required and this makes the overall cost higher. High temperature wells are used in flash steam power plants. In this method, water is injected into geothermal wells and this water gains heat from the earth. As a result, steam is produced and will run a steam cycle. These resources can be used in combined heat and power cycle if temperature is higher than 150°C.

ORC technology has widened the use of lower temperature wells. Nowadays, mostly medium temperature resources of 100 -220°C using binary technology have been utilized. In binary system, electricity production unit works in a separate cycle which is in heat transfer with the wells. As depicted in Figure 13, hot water from geothermal wells goes into hot side of evaporator and will heat the ORC fluid. Then ORC unit will produce power from this gained heat. Since the water from the well does not circulate in power unit this method has least pollution (The pollution is due to existence of toxic materials in water of geothermal wells)

Figure 13: Typical binary geothermal plant and its ORC unit
Utilization of lower geothermal wells of about 70 – 100 °C is under intensive investigation. Wells with these temperatures are available in many places of the world; hence, progress in using them will provide vast potential for geothermal based ORC units to produce power. To achieve this, effective and efficient heat exchangers should be used to minimize the heat losses.

Little temperature difference of less than 10 °C between forward and return lines in low temperature wells necessitates large heat exchangers area and high mass flow rates. These requirements make the system rather tough to be economically designed.

Since heat extraction from low temperature well needs high heat exchanger area, it would not be wrong to assume that most of the costs from low temperature binary power plants are due to heat exchangers. Having systems with efficient and optimum heat exchangers would be giant step forward in improving the cost effectiveness of system. Titanium flat plate heat exchangers which have good heat transfer and are compact and can be suggested (Kontoleontos E., Mendrinos D., Karytsas C, 2010). 

Regarding the working fluids, Isobutane (R600a) and R134a are recommended by Kontoleontos E., Mendrinos D., Karytsas C, (2010).

Due to recent progresses in ORC technology, geothermal ORC in low temperatures (about 70 – 100 °C) is economical nowadays. However some special techniques will help in making the system more economically viable. One novel idea is to use an absorption chiller in contact with cycle condenser. The idea is to use the ORC condenser as chiller evaporator. This will decrease the condenser temperature. Having less condenser temperature will give more temperatures difference between heat source and heat sink which results in better Carnot efficiency for ORC unit. This is easily proven: If Carnot efficiency is defined as temperature difference of heat source and heat sink divided by heat source temperature, it is clear that by fixed heat source temperature; less heat sink temperature connected to condenser will give better Carnot efficiencies.

Other ideas same as using Nano-structured metal-organic heat carriers can be helpful to enhance the ORC overall performance and reduce the ORC cost. This is a great area for research and development purposes. (Nathanael Baker, 2009).

4-2 Geothermal power from waste heat of oil and gas wells

Introducing low energy utilization power cycles will make it possible to produce geothermal power from oil and gas wells waste heat.

National Renewable Energy Laboratory in USA believes that these geopressed resources can be potentially used to produce 70 000 MW of electricity which accounts for 10% electricity consumption of whole country. Southern Methodist University is working on using produced hot water from oil and gas wells in united states to meet demands of some millions households.

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Using available oil and gas infrastructure make this option extremely interesting. Wells which are drilled for petroleum extraction normally have temperatures of above 100°C. Large amount of produced hot water from these wells which reach millions of liters are currently deemed waste. ORC units can produce electricity from waste heat in these hot waters from oil wells. Produced electricity can be used on site or be fed to grid. Normally the transmission infrastructure is already available near the wells and electricity can be transmitted easily.

One other source is depleted oil wells which are now abandoned. As a case in point it is estimated that oil and gas fields in Texas have potential waste heat to produce up to 2200 MW of geothermal power. Some demonstration projects have been run using geopressed resources from oil and gas fields in Texas and this is strongly believed that the technology will be well commercialized during next years. More research is ongoing on the subject.\textsuperscript{23}

5- Waste heat recovery applications of ORC

One new method of recovering waste heat in industries is to use low maintenance and safe ORC technology. It is known that most of processes have waste heat of less than 400-500°C which gives an abundant potential for ORC usage.

Supply of heat for absorption chillers in addition to electricity production with no more fuel consumption is the advantages of this power generation method. As a result all produced electricity from ORC waste heat recovery will contribute to reduce some emissions same as CO₂, SOₓ, etc, in an indirect way.

Trends for having more waste heat recovery units will be enhanced when more energy prices are in the scene. Moreover, cost evaluation of waste heat recovery systems are better to be investigated through their life cycle rather than their initial cost.

In this section waste heat recovery in steel, ceramic, and cement industries and also vehicles will be investigated. Waste heat recovery from exhaust gases of diesel engines have been studied in details by author in another project. A summary is brought here:

“There exist many options for obtaining better performance and more efficiency for engine based power generation. Wartsila is a leading manufacturing of internal combustion engines (ICE) used for power generation and is seeking various system options for increasing efficiency out of their plants. Project Group D of the Applied Energy Course Spring 2010 at KTH is tasked with exploring this topic for Wartsila. The company has already investigated into a few alternatives and the use of an Organic Rankine Cycle (ORC) is been deemed one such option. This sub-group of project group D is tasked with the optimization of an ORC cycle to be used as a bottoming cycle in a Wartsila’s ICE power plant design.

Organic Rankine Cycles are used for a wide variety of applications and various design options can be tailored to best optimize a certain scenario. A design and cycle optimization was carried out to fit the specifications given by the industry client. First, a research and literature survey was carried to explore various possibilities in designing an ORC. With this background and comprehension, various proposed layouts given by the client were then analyzed and modeled with the software Engineering Equation Solver (EES). After coming to conclusions about operating conditions and configuration for these proposed layouts, recommendations and suggestions were made in and to improve the overall ORC design.

We have found that although the Organic Rankine Cycle has the same principle of operation to the steam Rankine Cycle, it has many different considerations due to the special application of waste heat recovery. Our

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24 Gordon Denbow, Reza Rowshanazadeh, Andrew Tabije, Andrea Pla Rubio, Maria Chuvashova, Sara Ghaem, Amir Sohani, Adel Mikanik; available at: http://docs.google.com/fileview?id=0B5svs7AVkUW4NTk5M2UyMWYtZjk3NS00MzAxLWI4NzItODZlOTJjM2YxMjNl&hl=en&authkey=CLOMw-EI; as accessed: 07.07.2010.
research revealed that the most suitable choice for the ORC layout was the first layout operating at saturated conditions without a regenerator with an evaporator temperature of 140°C. Also it was found that using additional waste heat from engine cooling was more beneficial to the desired performance than, for example, incorporating a regenerator or changing system performance by the addition of superheat.

Also the working fluid in the cycle is very important and there are a few alternatives which actually have a higher thermodynamic efficiency for the same operating conditions, however may not be suitable for reasons of practicality; cost, toxicity and corrosiveness to components among other things. Since the power output of our ORC is in the range of 1MW (small scale), the best options with regard to turbine selection are different for large scale applications. The best current option for our capacity range was found to be the radial inflow turbine because of its high isentropic efficiency and flexibility at different operating conditions. However the Euler turbine shows great promise for future applications because of its lower cost of construction and robustness especially in vapour flows where moisture or contaminants may be present. Also the Euler turbine operates at lower speeds than the Radial Inflow turbine which decreases its size and the losses associated with operation; however the Euler turbine is still mostly at an experimental stage where its advantages have not yet been proven in practice. The ORC continues to be under investigation today since it utilizes waste heat that would normally just be dumped to the environment, this will continue to be important as the global situation with regard to energy and climate change intensifies. (Gordon Denbow, Reza Rowshanzadeh, Andrew Tabije,..(2010).)
5-1 Waste heat from Steel industry

In some countries same as Korea, up to 10% of total energy consumptions goes to Iron and steel industries. Nearly 40% - 50% of this amount is waste heat and only 15% of this waste heat is normally recovered.

In Sweden, the energy consumption per produced ton of steel is 17 GJ/t,. It is believed that this consumption can be decreased by 20-50%.25

Available waste heat in temperatures of less than 400°C makes ORC the most economical option produce power from this rather low grade heat. Proper efficiency, compactness and being economical are merits of ORC in this issue.

To categorize waste heat sources from steel industry, waste heat can be divided into two categories of clean and un-clean waste heats. Clean sources are normally from burning of methane and have low dust contents. These waste heats can be recovered by recovery heat exchangers and are normally from rolling and forging pre-heater furnaces. The second type of heat sources are so volatile in quantity and have high temperature and dust contents. This makes about 60% of total amount. Waste Heat recovery from them is under research and development. The second type can be found from fumes of blast and electric arc furnaces.26

5-1-1 Exergy analysis

Before 1970s energy analysis was best method to analyze a thermodynamic process. After that and a trend to reduce impact and losses of energy conversion methods to environment, minimization of entropy or exergy analysis has come seriously into play. As wanted, this has helped researchers to reduce internal irreversibility of processes and their losses to environment.

Moreover, First law of thermodynamics deals with quantities while quality of energy is not considered by that; hence, exergy analysis is of enormous importance nowadays. This will help recognizing the potential for work extraction from a heat source to the point it reaches equilibrium with the ambient. This method will give better view on efficiency of each thermodynamic process and provide information on both quality and quantity of a heat source.

For constant pressure solid or liquid with known heat flow the exergy term is defined as:
\[ \varphi = Q \left( 1 - \frac{T_0}{T} \ln \frac{T}{T_0} \right) \]

\( T_0 \) stands for minimum cycle temperature where the heat can be dumped there and \( T \) is process temperature.\(^{24}\)

Typical waste heat of Iron and steel production site is shown as Figure 14.

![Figure 14: Steel industry typical waste heat sources and their potential](source: S. Malmström\(^{27}\))

As depicted, some cases same as cooling water have high amount of energy, but since the temperature is low the quality will be low as well and this heat cannot be so useful for ORC applications. Surface losses are not focused in some locations and it is hard to harness them during operation without interrupting the process.

Among the mentioned waste heat sources in the Figure 15, waste heat from slag have high potential for ORC applications. Power can be extracted from this source without interruptions of the process. However, to use this source, heat transfer from solid material shall be considered and collecting and managing the waste is also important and difficult. It is a case which is under research.

Steam from hot coke cooling is also a difficult source to deal with. Alternative cooling methods might make a proper heat source from hot coke cooling medium. Assuming that corrosion problems during flue gas condensation can be solved by appropriate choice of material and design, then the waste heat from flue gas is another promising option. Flare gas can be used if the gas is not needed for any other applications (S. Malmström, 2009). 23
5-2 Ceramic industry

Ceramic industry as not highly efficient technology has abundant of waste heat. ORC can generate electricity from these sources and the produced power can be used in exhaust fans and to circulate air.

The most prone spot in the industry is the flue gasses of Kiln. The efficiency of installed ORC depends on exhaust gas temperature and varies between 8 to 11 %.

Based on 7000 and 8000 operation hours internal rate of return will be about 0.18% and payback time will be about 3-4 years.

5-3 Cement industry

One of high endothermic processes called de-carbonization in cement industry requires hot heat supplies. ORC can use waste heat of heat this reaction, combustion gas used for preheating of raw material or cooler air stream used to cool clinker.

Normal cement industries use 3-5 GJ of heat per ton of clinker. From waste heat of processes, nearly 1MW electricity per 1000 ton clinker can be produced. 2000 -8000 ton clinker can be normally produced per day from a cement plant and this indicates the great potential of ORC based power generation in this industry. Up to 20 % electricity consumption of this industry can be generated via ORC application using its own waste heat as heat source (Riccardo Vescovo, 2009). 

In Figure 15 (Turboden, 2009) a cost and performance study is used to give an overview on effectiveness of ORC in waste heat recovery options for three cases of steel, cement and float glass industries. Some terms have been defined in section A-2. As it is seen in the figure, for all three plants, internal rate of return is above 20% and payback time is less than 5 years. These and greatly positive net present value show that all option are economically interesting. In addition to that, it is calculated that for each industry, about 10’000 tons of CO2 per year can be avoided and this environmental benefit is added to its economical profits.

---

<table>
<thead>
<tr>
<th>Industry/application</th>
<th>Cement</th>
<th>Float glass</th>
<th>Steel flat products (rolling mill)</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat source</td>
<td>Kiln and clinker cooler gas</td>
<td>Oven exhaust gas</td>
<td>Preheating oven exhaust gas</td>
<td></td>
</tr>
<tr>
<td>Plant capacity</td>
<td>2,500</td>
<td>500</td>
<td>6,000</td>
<td>Tons per day</td>
</tr>
<tr>
<td>Electricity cost a</td>
<td>0.09</td>
<td>0.095</td>
<td>0.06</td>
<td>€/kWh</td>
</tr>
<tr>
<td>Wasted thermal power in exhaust gas b</td>
<td>12</td>
<td>5</td>
<td>13</td>
<td>MW</td>
</tr>
<tr>
<td>Thermal power to ORC</td>
<td>11</td>
<td>4.7</td>
<td>13</td>
<td>MW</td>
</tr>
<tr>
<td>Thermal power to thermal users</td>
<td>1</td>
<td>0.3</td>
<td>0</td>
<td>MW</td>
</tr>
<tr>
<td>Net ORC electric production</td>
<td>1.6</td>
<td>1</td>
<td>2.4</td>
<td>MW</td>
</tr>
<tr>
<td>Net electricity production c</td>
<td>12,000</td>
<td>8,000</td>
<td>19,200</td>
<td>MWh/y</td>
</tr>
</tbody>
</table>

**Capital expenditure indications**

<table>
<thead>
<tr>
<th></th>
<th>Cement</th>
<th>Float glass</th>
<th>Steel flat products (rolling mill)</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>ORC cost</td>
<td>1.8</td>
<td>1.3</td>
<td>2.4</td>
<td>Million €</td>
</tr>
<tr>
<td>Balance of plant d</td>
<td>2.6</td>
<td>1.1</td>
<td>1.5</td>
<td>Million €</td>
</tr>
<tr>
<td>Total cost (+10% project management)</td>
<td>4.8</td>
<td>2.6</td>
<td>4.3</td>
<td>Million €</td>
</tr>
</tbody>
</table>

**Annual Cash flows d**

<table>
<thead>
<tr>
<th></th>
<th>Cement</th>
<th>Float glass</th>
<th>Steel flat products (rolling mill)</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operational expenditure</td>
<td>-40,000</td>
<td>-40,000</td>
<td>-40,000</td>
<td>€/y</td>
</tr>
<tr>
<td>Cash flow – electricity</td>
<td>1,152,000</td>
<td>760,000</td>
<td>1,152,000</td>
<td>€/y</td>
</tr>
<tr>
<td>Cash flow – heat e</td>
<td>240,000</td>
<td>72,000</td>
<td>0</td>
<td>€/y</td>
</tr>
<tr>
<td>Net cash flow</td>
<td>1,392,000</td>
<td>832,000</td>
<td>1,222,000</td>
<td>€/y</td>
</tr>
</tbody>
</table>

**Results f**

<table>
<thead>
<tr>
<th></th>
<th>Cement</th>
<th>Float glass</th>
<th>Steel flat products (rolling mill)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Profit before tax</td>
<td>4</td>
<td>3.7</td>
<td>4.4</td>
</tr>
<tr>
<td>Internal rate of return (10 years)</td>
<td>25%</td>
<td>27%</td>
<td>23%</td>
</tr>
<tr>
<td>Net present value (10 years)</td>
<td>€5,333,129</td>
<td>€3,310,109</td>
<td>€4,091,971</td>
</tr>
<tr>
<td>Avoided CO₂ emissions g</td>
<td>9,664</td>
<td>5,520</td>
<td>12,096</td>
</tr>
</tbody>
</table>

**Notes:**

a. These values include incentives f any; differences are due to total power installed, nation etc.
b. Assuming to cool down the gas to 150/160°C
c. Including heat recovery exchanges and civil works – estimated by reputable suppliers
d. Assuming 8,000 operating hours/year
e. Assuming a heat valorization of circa 0.03 kWh
f. Assuming discount rate of 5%
g. Assuming 0.63 kg of CO₂/kWh electric and 0.2 kg of CO₂/kWh thermal (from CH₄ combustion).

**Figure 15: Cost effective application of ORC in three waste heat recovery applications**
Source: Turboden, Brescia, Italy

5-4 ORC in vehicles

Exhaust gas and coolants of vehicles can be used to generate power by ORC. For normal cars in cruising speed this will produce 10% more power and can be used to recharge batteries of a hybrid car and improve its overall efficiency. In light duty cars depending on type this can cause between 5-30% reductions in fuel consumption. BMW, Honda and Researchers at Loughborough University and the University of Sussex, UK are examples of researchers in this field. 30 “Their test results were presented in 2008. It shows that 62 miles/h speed of driving; using ORC will improved the thermal efficiency of the engine by 3.8%”. (greencarcongress, 2009).

As depicted in Figure 16, ORC unit with evaporator, expander, condenser, and pumps are used to produce extra amount of power in cars. Evaporator gains heat from waste heat of engine while the condenser is connected to its radiator. Expander and generator will produce electricity. Then the power is stored in batteries. This will help the engine to work in more efficient ways by using extra amount of produced power.

![Figure 16: ORC in vehicles](image)


Opcon in Sweden is also installing ORC boxes using Lysholm turbine on ships. It is claimed that it will cause fuel saving of about 4-6%. Shipping accounts for 5% CO₂ emission and produces MWs of power which a fraction is deemed waste. 32

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6- ORC as bottoming cycle in combined power cycles

In this section, ORC as bottoming cycle of gas turbine with lower exhaust temperature than usual will be discussed. The gas turbine as top cycle can be recuperative gas turbine or a gas turbine with large pressure ratios and low temperature exhaust.

Normally the bottoming cycle of gas turbine is Rankine cycle. This cycle have good combination with gas turbine, high efficiency and well known equipment and processes.

If the gas turbine exhaust temperature is rather low, ORC and Kalina cycle will be selected. Since Kalina is not economically viable now, ORC will be the first selection since they are currently commercially obtainable.

One significant advantages of ORC and gas turbine is that they can compete with high temperature exhaust gas turbine even if the temperature is lower, hence this will give the opportunity to have gas turbines with lower turbine inlet temperature, less combustion temperature and consequently less NOx production, manufacturing and operational cost. Gas turbine will be cheaper since less resistant material and cooling methods will be necessary.

Regarding the working fluid, it should be a dry one (See section A-2 for definition of dry fluid), stable at related temperature range and meet the limitations of minimum and maximum cycle pressures. Toluene and Cyclohexane are among best options in this special application of ORC since they give rather proper efficiency. Toluene will give better heat recovery in economizer due to closer temperature line with heat source and gains a little more efficiency. Cyclohexane is less sensitive to top cycle temperatures, but is a bit more expensive and needs larger heat recovery vapor generator which is the most expensive part in the cycle. Security to control the flammability and toxicity of these working fluids shall be considered carefully although bottom cycles maximum temperature should be far from ignition point of these fluids (R. Chacartegui, J.M. Muñoz, T. Sánchez, 2009).

Considering less than 500$/KW for gas turbine and 2000$/KWe for ORC, electricity price of 100$/MW and fuel cost of 38$/MW the payback time will be less than 7 years and internal rate of return is more than 0.15 which makes the system economically feasible. The calculated values are related to a recuperated gas turbine with top cycle and bottom cycle power ratios of about 3 and inlet temperature of about 1000°C for gas turbine. More ORC or gas turbine prices will lead to longer pay back time and less internal rate of return, as a result, produced electricity price cannot compete with conventional cycles. 33

In Table 4 results from a study30 (R. Chacartegui, J.M. Muñoz, T. Sánchez, 2009) are brought to show in which commercial gas turbines ORC combination will give acceptable efficiencies and performance. Here, power ratio is the ratio between bottoming cycle and the whole cycle power and is presented in percentage. It

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is shown that for lower temperature gas turbines ORC can well compete with steam cycle, even if multiple pressures heat recovery steam generator is used. The compared working fluids with water were, CHEX(=cyclohexane), R113, and Toluene. It is concluded that CE LM-6000 dry, Rolls-Royce WR 21 and Solar Mercury models gas turbine models with less exhaust gas temperature are suitable for ORC application (R. Chacartegui, . Sánchez, J.M. Muñoz, T. Sánchez 2009).

<table>
<thead>
<tr>
<th>Efficiency/power ratio (%)</th>
<th>Toluene</th>
<th>CHEX</th>
<th>R113</th>
<th>Steam</th>
</tr>
</thead>
<tbody>
<tr>
<td>General electric PG-7251</td>
<td>54.9/32.7</td>
<td>53.4/30.8</td>
<td>49.9/26</td>
<td>55.1/33</td>
</tr>
<tr>
<td>Siemens V 94.3A</td>
<td>55.8/31.5</td>
<td>54.3/29.6</td>
<td>50.9/24.9</td>
<td>55.4/31</td>
</tr>
<tr>
<td>Alstom GT 24/26</td>
<td>55.1/30.7</td>
<td>53.7/28.9</td>
<td>50.4/24.2</td>
<td>55.5/31.2</td>
</tr>
<tr>
<td>Mitsubishi M701 G</td>
<td>56.3/29.8</td>
<td>54.9/28</td>
<td>51.7/23.5</td>
<td>56/29.4</td>
</tr>
<tr>
<td>GE LM-6000 dry</td>
<td>57.8/27.6</td>
<td>55.6/26</td>
<td>53.5/21.9</td>
<td>54.2/22.9</td>
</tr>
<tr>
<td>Rolls-Royce WR 21</td>
<td>53.3/22.7</td>
<td>52.2/21</td>
<td>50/17.6</td>
<td>48.1/14.4</td>
</tr>
<tr>
<td>Solar Mercury 50</td>
<td>54.9/29.8</td>
<td>53.4/27.9</td>
<td>50.4/23.6</td>
<td>48.9/21.3</td>
</tr>
</tbody>
</table>

Table 4: ORC in comparison with steam cycle for gas turbine bottoming cycle


It is also worthwhile that ORC as bottoming cycle for micro turbines of less than 500KW is discussed in “Micro scale applications of ORC down to 1kW” section.
7- Waste heat recovery in fuel cells

This is true that fuel cells have larger initial investment than conventional internal combustion engines, gas turbines and steam cycles, but it can provide cleaner energy conversion in some applications and in long term is able to compete with those technologies if net present value of the system is considered. (For more information about fuel cell please refer to appendix A-2).

In many rather high temperature fuel cells, considerable share of input energy is converted to unused heat. This heat leaves the fuel cell via cooling medium or exhaust gas. There are several methods which this waste heat can be recovered and consequently bettered efficiencies and electricity price will be achieved.

Fuel cell can have a bottoming cycle same as Brayton gas, thermoelectric, and organic Rankine cycle. Moreover, fuel cells waste heat can be used in fuel reforming processes, combined heat and power or Trigeneration plants.
7-1 Gas turbines as bottoming cycle

Gas turbines are one of options for utilizing not burned fuel and waste heat from Solid Oxide fuel cells. The system normally consists of Solid Oxide fuel cell internal gas reforming stack, compressors, recuperator, a combustor and gas turbines. In figure 16 a layout of system has been shown. As depicted in the figure 17, exhaust gasses of SOFC will be added to gas turbine combustion chamber. After producing high temperature gasses, turbines will be used to produce extra amount of power. Even Recuperators which pre-heat the entering air of compressors can be added to the system to utilize much more waste heat from SOFC.

![Diagram](image)

**Figure 17: Gas turbine as bottoming cycle of fuel cell**

Source: Journal of Power Sources

If a heat recovery system for cogeneration is added, up to 80% efficiency will be gained, otherwise, around 60% efficiency will be the result. The cost of a typical micro gas turbine in conjunction with Solid Oxide fuel cell will be around 2500$/kWe with about 50% efficiency. More efficient systems will cost more. Some modifications same as water injection to combustion chamber will help in increasing the efficiency.

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7-2 ORC as bottoming cycle of Solid oxide fuel cell

In Molten Carbonate Fuel Cells, large amount of heat is generated. Sensible heat from this waste heat can only potentially produce 25% more electricity. The amount of this waste heat is more than 30% of input energy. Even more amount of waste heat is available for Solid Oxide fuel cells.

Exploring some bottoming cycles, it is known that steam cycles are only economically viable at large scales. An alternative would be ORC. By applying ORC, as an example, Solid Oxide fuel cell total efficiency will increase by amount of about 15%. Even a single effect absorption chiller can be connected to Solid Oxide fuel cell and ORC while the waste heat of ORC is used for heating purposes and the rest for driving the chillers. This will help in another 20 percent increase while heat, cooling and electricity are produced (Fahad A. Al-Sulaimana, Ibrahim Dincerb, Feridun Hamdullahpurc, 2010).

In figure 18 a possible layout of the system is shown. In this figure, waste heat of SOFC which has temperatures of about 400°C will be transferred to ORC evaporator to heat the ORC fluid. Evaporated fluid will run the ORC turbine to produce extra amount of power. Efficiency of SOFC is about 60% and produced power from each SOFC unit can be about 100 KW. By adding ORC, more power and better efficiencies will be achieved. Waste heat of ORC condenser can be used in absorber chiller to produce cooling and add to total system efficiency as well (Fahad A. Al-Sulaimana, Ibrahim Dincerb, Feridun Hamdullahpurc, 2010).

It is assumed that cost of produced electricity after adding ORC is about 2500$/kW and can be less in larger plants. Using ORC will reduce the total cost per kW and in the same time will improve efficiency. There will be more motivation for ORC in lower exhaust temperatures of fuel cell in comparison with gas turbine applications. This is mainly true for Molten Carbonate Fuel cells. Mixture of fluids especially Siloxanes seem to be best options for working fluids (Wamei Lin, 2010).

Simplicity, rather proper efficiency in temperature range of about 100 – 400°C has made ORC one of best options for molten carbonate fuel cells bottoming cycle. Condenser pressure of less than 1 bar and up to 10 bars helps in selecting the turbine size cost effectively. Moreover, ORC cycle can be optimized depending on heat source and by changing best working medium. This means flexibility which is another advantage of using ORC.
7-3 Thermo-electric power generation

In solid oxide fuel cells another method called thermoelectric can be used as waste heat recovery option. This technology can bring down the cost of produced electricity to about 400$/KW which is an interesting amount and well close to targets for fuel cell technology. The efficiency of system can be over 60% in this configuration. The system is low maintenance, has low noise level and is a modular option to be combined with Solid Oxide fuel cells with 400-800°C waste heat. This system is so cost effective and coal can be used as a fuel for this system in a rather clean way.  

7-4 Fuel reforming by waste heat

Polymer electrolyte membrane fuel cells (PEMFC) have been recently well developed and are so interesting for transportation applications. It is hard to prepare hydrogen which is their main fuel. One method is to utilize waste heat of Solid Oxide fuel cells to reform some fuels to be used in PEMFC.

A layout of system has been shown in Figure 19. As depicted in the figure, waste heat of solid oxide fuel cell will be used in shifter in present of a catalyst to produce H\textsubscript{2} from added water and O\textsubscript{2} to SOFC exhaust gasses. This produced H\textsubscript{2} can be used in PEMFC for producing extra amount of electricity. Even after PEMFC, small fraction of waste heat will generate some amount of steam for further use (Wamei Lin,2010).

\begin{figure}[h]
\centering
\includegraphics[width=0.7\textwidth]{Figure19.png}
\caption{Waste heat recovery from Solid Oxide fuel cell to reform fuel for Polymer electrolyte membrane fuel cells.}
\end{figure}

Source: Infoscience\textsuperscript{38} [7]


Some water will be added in reformation process; hence this will help in producing more \( \text{H}_2 \). Normally hydrogen comes from fuel and here from water as well. The combine system has 20\% less cost than Solid Oxide fuel cell and 30\% less cost than PEMFCs. The efficiency of PEMFC will also be improved from about 40\% to 60\%.

7-5 Cogeneration arrangement of fuel cell

Using low temperature waste heat of some fuel cells same as Phosphoric Acid Fuel Cells can be used for heating purposes. This waste heat is not appropriate for gas turbine or ORC applications to produce power.

To sum up, depending on fuel cell technology waste heat recovery system can be opted. In case of Solid Oxide fuel cell gas turbine or micro gas turbine can be added as bottoming cycle. If PEMFCs are interested Solid Oxide fuel cell waste heat can be used to reform fuel for them. Cost effective Thermo electric systems or Trigeneration plant based on ORC and Solid Oxide fuel cells can also be considered. This will help to get high efficiencies and to produce electricity, heat and cooling simultaneously. This is while that, in case of Molten Oxide fuel cells ORC is very interesting option. Low temperature fuel cells waste heat can only be used to meet cooling or heat demand. Cost of added technology is normally less than the fuel cell itself and efficiency will be improved by this extra system. This is true for fuel cell based ORCs as an option.

Aforementioned conclusion is summarized in Table:

<table>
<thead>
<tr>
<th>Fuel cell type</th>
<th>Heat recovery options</th>
<th>Estimated cost of produced electricity after heat recovery</th>
<th>Best recommended option</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOFC</td>
<td>Gas turbine, ORC, TE as bottoming cycle, Reforming fuel for PEMFC</td>
<td>Claimed to be about 400$/kW</td>
<td>Thermal electric or Trigeneration system based on ORC</td>
</tr>
<tr>
<td>MOFC</td>
<td>ORC</td>
<td>2500$/kW</td>
<td>ORC</td>
</tr>
<tr>
<td>PEMFC</td>
<td>No significant heat recovery option</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5: Heat recovery options from fuel cells
8- Gas-cooled nuclear power plant with closed Brayton cycle combined with ORC

Nowadays, more interest for cost effective, fuel efficient and safer reactors has caused investigation on helium-cooled nuclear reactors. The gas turbine-modular helium reactor (GT-MHR) is one type of helium-cooled nuclear reactors (Mortaza Yari, S.M.S. Mahmoudi, 2010). These reactors are designed to be connected to a closed Brayton cycle.

As depicted in Figure 20, after that helium works as a coolant of reactor core, it poses enough temperature (point 1) to run a gas turbine cycle in order to produce electricity. This gas turbine cycle has turbine (process 1-2), a low pressure (process 5-6) and high pressure compressor (process 8-9). The compression occurs in two stages in order to decrease input power to compressors.

Helium after intercooler between low pressure and high pressure compressors (Point 8) shall be cooled down to 26 °C (between two stages of compression in the cycle). This cooling process has about 300 MWth waste heats provided 100 – 150°C temperature at inlet of intercooler. Moreover, pre-cooler of helium (Process 5-6) before low pressure compressor has an amount of waste heat. Intercooler and pre cooler waste heat make enough justifications for the ORC usage in the cycle. In Figure 19 it has been shown how two ORC units will fit in the cycle. One ORC evaporator will use intercooler and the other use pre-cooler waste heat to produce extra amount of power. Adding ORC to the system will decrease the exergy loss of system about 5% and will improve the efficiency by about 3% (Mortaza Yari, S.M.S. Mahmoudi, 2010).

Figure 20: ORC usage in Gas-cooled nuclear power plant with closed Brayton cycle.
9- Solar thermal electricity generation

During last 20 years, solar thermal based electricity generation has been significantly increased globally. Nowadays, just 1% of market share of ORC is related to this technology. In here bellow part, benefits and huge potential of this technology for ORC applications will be introduced.

It is generally believed that higher temperature and heat for power cycle yields better performance, as a result steam cycle using water is normally used in solar thermal electricity production systems and attempts have been done to achieve higher temperatures from solar collectors. However, looking more precisely to the system will show several justifications for using ORC. First, turbo machineries for steam cycle are not so efficient in related rather low temperatures of solar systems in comparison with conventional systems. More importantly, to avoid condensation during expansion in steam turbine superheating to temperatures of 600°C is necessary. These temperatures for solar thermal systems are so expensive to be achieved. Thermal energy storage is not even commercialized in these temperatures and solar detectors and high concentrator equipments are expensive in these cases. This is while using ORC instead of steam Rankine cycle does not have these drawbacks.

No freezing of working fluid during cold weather threatens ORC fluid while this problem exists in steam cycle which uses water. Since in ORC less heat source temperature in comparison with steam cycle is used, less costly material for solar collector and concentrators is not needed and this will reduce the total cost of system in comparison with steam cycle (Pei Gang, Li Jing, Ji Jie, 2010). This system can be combined with phase change material (PCM) storage system in relatively reasonable costs to increase the period of power generation. In next part, brief description from a sample ORC based solar thermal system is given.

9-1 Solar thermal electric production system with PCM storage

In this system (See Figure 21), solar collectors will receive energy from sun. The gathered heat is transferred to evaporator of ORC unit or it will be stored in phase change material storage system. Overall layout of the system is illustrated in Figure 20. There are three mode of operation for system:

1- When sun energy is available and demand for electricity is high: No storage will be done and collector heat will be used in ORC evaporator to evaporate the ORC fluid and run the turbine, consequently, electricity will be produced in ORC unit.

2- When energy from sun is available but demand is not high: In this case heat from solar collector is stored in PCM storage.

3- When energy from sun is not available and electricity demand exists: Stored energy in PCM storage is used (In this storage system, storage material will change phase by gaining heat and when necessary it will give the heat back by returning to initial phase.)
In case (1) valves with numbers of 1, 4, 5, 6 (V1, V4, V5 and V6) will be open to produce electricity in ORC unit and storage is not done.

In case 2, by closing pumps 1, 2 and opening 2, 4, 5 valves, the heat will be stored in PCM storage system.
In case 3, the heat from storage system will be used by opening valves 1 and 3.

One other great potential application of solar ORCs is in small scale remote power systems. This new technology is discussed in “Micro scale application of ORC down to 1KW” section. Moreover, solar based ORC in small scales can be coupled with reverse osmosis desalination unit which will be discussed in that section as well.

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9-2 Cost of Solar based ORC

Cost of this system is quiet lower than photovoltaic cells. Least price for PV technology has been assumed to be 8000$/KW (Solarbuzz, 2010)\(^{41}\) while the storage is not easily done. This is while that in solar based ORC systems, on average 2000$/KW \(^{55}\) should be considered for ORC and adding 200$ per square meter of solar field will yield in quiet less price than other solar based technologies same as PV. Here, it is conservatively assumed that on average each 8 m\(^2\) of solar collector will be capable of 1 kW electricity generation (J. Bruno, J.-Villada, E.Letelier, S.Romera, A.Coronas, 2007).\(^{55}\)

Development of ORC based concentrated solar power technology in the market suffers from lack of awareness while more attention is on photovoltaic cells. \(^{42}\) Based on a study (J. Bruno, J.-Villada, E.Letelier, S.Romera, A.Coronas, 2007) cost of a solar based ORC and desalination unit is 40-60% less in comparison with PV and desalination unit.\(^{55}\)

Optimization of solar based ORC will be discussed in “Optimization of Micro ORC as CHP or CCHP system based on solar energy” chapter.

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\(^{41}\) Solarbuzz; available at: [http://www.solarbuzz.com/StatsCosts.htm](http://www.solarbuzz.com/StatsCosts.htm); as accessed: 10.10.2010.

II- In depth study:

1- More detailed case studies – Cost reduction and performance improvement of ORC

In this section, it is aimed to optimize ORCs for thermal efficiency, net output power, exergetic efficiency, total necessary heat transfer area and cost. As a result this section deals with more details while previous parts included more general aspect of ORC applications.

To reach aforementioned goal, equipment of ORC shall be selected properly. The working fluid, ORC turbo machineries speed, degree of superheat, working minimum and maximum temperatures and pressures shall be selected properly.

A model for thermal and exergy efficiencies, heat transfers factor and turbine size has been built in EES. The outputs give information on efficiency and cost of cycle as some numbers. Then, effect of selecting components with different characteristics and ORC features has been studied on them. To assure the accuracy of the model, some results have been checked with available information in literature which will be mentioned in next sections.

Before starting to study the system in details, it is interesting to calculate the normal ORC Carnot efficiency. This will give us a good estimate on maximum theoretical efficiency which the cycle can gain and to compare the gained thermal efficiencies with that.

Carnot efficiency is the temperature difference between heat source and heat sink divided by heat source temperature (in Kelvin). For an ORC cycle with heat Source temperature of 400K and heat sink temperature of 300K, the Carnot efficiency will be about 25%. It is quiet lower than Carnot efficiencies for a normal steam cycle. As a result, this figure should be kept in mind to prohibit down grading the ORC performance while its thermal efficiency is considered.
1-1 Effect of superheating on ORC performance

Normally it is true that having more inlet turbine temperature means more cycle efficiency. In some cycles with wet fluid (Refer to A2- definition of terms), super heating is essential in order to keep the expansion in vapor phase and to prevent wet expansion and erosion in turbine. Since most ORCs use dry fluids, having excessive superheating is proven to have negative effect on maximum working pressure, saturation temperature and performance of the cycle. This is shown in Yiping Dai, Jiangfeng Wang, Lin Gao (2009). To reassess their findings and to check consistency of established EES model with previous findings, result of investigation is plotted and shown in Figure 22. This figure is in good consistency with previous findings and indicates that excessive superheating have negative effect on total efficiency of ORC. (Please refer to appendix for definition of Dry, wet, isentropic fluids and pinch point temperature).

For calculating total efficiency (thermal efficiency) in the model, equation 1 is used. \( W_{net} \) means net produced power from cycle and \( Q_{eva} \) stands for heat input from heat source to evaporator.

**Equation 1:**

\[
\eta_{total} = \frac{W_{net}}{Q_{eva}}
\]

![Figure 22: Superheating effect on total efficiency and work output](image)

---

In addition to reasons provided in mentioned articles and the given data from EES, regarding superheat it should be noted that: available heat source for ORC can be used in both evaporator and superheater. If more fraction of heat is used in evaporator, higher temperature and pressure in evaporator will be achieved (More heat means more heat transfer and this increase evaporator temperature and pressure). Since evaporator pressure above 20 bars and high temperatures will raise the cost of material and safety measures in this heat exchanger, if high pressure and temperature is reached in evaporator, more fraction of input heat will be used in super heater. Otherwise, it is interesting to limit the superheat to increase ORC efficiency (as mentioned by and Yiping Dai, Jiangfeng Wang, Lin Gao, 2009 and gained by EES model). This can be done by giving the right amount of heat from heat source to either evaporator or superheater and also choosing the right pressure increase by pump. For adjusting pump pressure see chapters II-1-6.

In next step, advantages of using rather no superheat which is possible for ORC cycles will be discussed. Due to using dry or isentropic fluid in ORC, dry expansion in turbine is possible and no or low super heat case is practical. In Table 5 some benefits of using no superheat and even admission of wet fluid is summarized. Wet fluid admission is possible in hermetic expanders and Euler turbines which will be discussed in “Turbine selection”, 10-4, section. (Definition of terms “Turbine” and “Expander” are given in section A-2).

<table>
<thead>
<tr>
<th>Items to be benefited</th>
<th>Turbine</th>
<th>Boilers</th>
<th>Other heat exchangers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reasons for cost reduction</td>
<td>Lubrication</td>
<td>Improved power out put</td>
<td>-Separator and external circulation line is not needed</td>
</tr>
<tr>
<td></td>
<td>- No need for separate lubrication system including oil separator, pump, storage tank, timing gear and shaft seal.</td>
<td>Removal of superheat, Less heat is needed</td>
<td>-Boilers with cheaper type of plate fin</td>
</tr>
<tr>
<td></td>
<td>- Possibility for mixing oil with working fluid and an extraction line after feed pump to turbine bearings (Fluid will be used for lubrication)</td>
<td>no need for de-superheat means</td>
<td>- Most importantly, no superheat means higher evaporator temperature with the same heat source and this will increase the total efficiency</td>
</tr>
<tr>
<td></td>
<td>Means better efficiency for dry and isentropic fluids</td>
<td>more enthalpy drop potential after turbine and this means more power</td>
<td></td>
</tr>
</tbody>
</table>

Table 5: Benefits of wet fluid admission to turbine

Source: ASME, H. Leibowitz, I. K. Smith, N Stosic, 2006. 52

To clarify items of table 5, it is evident that if superheating is omitted, less input heat in the cycle is needed since less temperature after evaporator is needed. De-superheat after expansion will be removed, since no
superheating is done and fluid after expansion is closer to saturation vapor state. As already mentioned, according to Yiping Dai, Jiangfeng Wang, Lin Gao (2009), and provided model, with dry and isentropic fluids less superheats means better efficiencies. Other items in Table 5 related to boilers and heat exchangers will be discussed in their respective section of 1-8.

A study (H. Leibowitz, I. K. Smith, N Stosic, 2006) shows that typical 24kW electricity ORC with R245fa would be best efficient when admitted fluid to turbine is 88% in quality and this is possible with hermetic expanders. Normal air compressor with 4600 rpm and 102mm male diameter in reverse mode of rotation can be used as expander with about 90°C to give best efficiency.

1-2 Exergy losses and efficiency

For developing the computer model in EES, exergy losses of components and exergy efficiency is added. For this reason, equations 2, 3 and 4 are used. Exergy of each state in cycle is denoted by e, enthalpy by h and standard condition which is 25°C temperature and 1 bar pressure is showed by 0 subscripts:

**Equation 2:** \[ e = h - h_0 - T_0(S - S_0) \]

Exergy for different states of ORC cycle is calculated. Exergy loss in each component is defined as equation 3. In this equation, exeloss stands for exergy loss in component, \( e_{in} \) and \( e_{out} \) show the inlet and outlet exergy of each component. W shows the work which is done by that component (For heat exchangers it is assumed to be zero). In equation 4, exergy efficiency is shown by \( \eta_{exe} \). This efficiency shows the used exergy in total cycle divided by input exergy of the cycle. Total here means summation for all components of cycle.

**Equation 3:** \[ exeloss = e_{in} - e_{out} - W \]

**Equation 4:** \[ \eta_{exe} = 1 - \frac{exeloss_{total}}{e_{in total}} \]

In the generated computer model and coming tables, \( exeloss1 \) means exergy loss in evaporator. \( Exeloss2 \) means exergy loss in turbine and \( exeloss3 \) stands for exergy loss in condenser.

By using aforementioned equations in the model and running that for ORC cycle, results as in Table 6 shows that by adding the pinch point temperature the exergy loss in the evaporator (exeloss1) will increase.
Table 6: Effect of pinch point temperature on exergy loss in evaporator (exeloss1)

By having different setups in EES model, it was concluded that for ORC plants with more output power and higher input temperature, most of exergy loss occurs in evaporator. That is while that for smaller plants with less heat source temperature exergy loss of turbine will be the most dominant one (exergy loss 3). These results are concluded from Table 7 with a rather high power cycle and Table 7 with rather lower heat source ORC.

By comparing several articles (e.g.: Wang \(^{44}\), Tchanche \(^{45}\)) these results can be confirmed as well. Comparing situations in these articles shows, in solar desalination unit with low powers most of exergy loss comes from turbine, but in cement industry with more ORC power, evaporators are the spots with most of exergy losses. Furthermore, it is known that for small scale applications about 90% efficient heat exchangers \(^{46}\) and for larger scale ORCs, efficient turbines are in the market now. This confirms the above results as well.

<table>
<thead>
<tr>
<th>Run</th>
<th>(\delta_T)</th>
<th>Tpp</th>
<th>exeloss1</th>
<th>(\eta_{\text{total}})</th>
<th>exeloss2</th>
<th>Q_t [kW]</th>
<th>T_{1aug}</th>
<th>T_{1sat}</th>
<th>Wret</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.903</td>
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<td>3.112</td>
<td>0.01221</td>
<td>5.713</td>
<td>3.658</td>
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<td>95</td>
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<td>3.647</td>
<td>3500</td>
<td>90.94</td>
<td>90</td>
</tr>
<tr>
<td>3</td>
<td>0.1604</td>
<td>20</td>
<td>5.658</td>
<td>0.002118</td>
<td>6.204</td>
<td>3.626</td>
<td>3500</td>
<td>85.16</td>
<td>85</td>
</tr>
</tbody>
</table>

Table 7: Low heat source temperature ORC (ORC cycles with heat source temperature of mostly less than 150°C).

In Table 9, it is also shown that by increasing the pinch point temperature, evaporator exergy loss (exeloss 1) will be increased and this will make its exergy loss even higher than in turbine.

It is also concluded that same trend as thermal efficiency for exergy efficiency exists. With more degree of superheat both efficiencies will go down. This is concluded from Table 9.

<table>
<thead>
<tr>
<th>(\delta_T)</th>
<th>Tpp</th>
<th>exeloss1</th>
<th>exeloss2</th>
<th>(\eta_{\text{ex}})</th>
<th>exeloss3</th>
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</thead>
<tbody>
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<td>21.19</td>
<td>5</td>
<td>41.98</td>
<td>9.361</td>
<td>0.7612</td>
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</tr>
<tr>
<td>23.48</td>
<td>10</td>
<td>42.27</td>
<td>9.059</td>
<td>0.7764</td>
<td>7.618</td>
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<tr>
<td>25.97</td>
<td>15</td>
<td>43.54</td>
<td>8.741</td>
<td>0.7713</td>
<td>8.01</td>
</tr>
<tr>
<td>28.67</td>
<td>20</td>
<td>44.9</td>
<td>8.406</td>
<td>0.7658</td>
<td>8.408</td>
</tr>
<tr>
<td>31.56</td>
<td>25</td>
<td>46.35</td>
<td>8.055</td>
<td>0.7601</td>
<td>8.813</td>
</tr>
</tbody>
</table>

Table 8: Exergy efficiency changes with degree of superheat

\(^{44}\) Jiangfeng Wang, Yiping Dai, Lin Gao, Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry, Applied Energy 2009; 86: 946.


\(^{46}\) AIREC; available at: http://www.airec.se/; as accessed: 17.06.2009.
1-3 ORC speed

In order to keep ORC small in size, safe in operation and efficient, right speed for ORC shall be selected. In some cases the turbine, pump and generator are on the same shaft and they rotate in same speed. In this case, ORC speed means pump, turbine and generator rotational speed.

Not optimizing the speed in an ORC may lead to damages to seals of turbine, limitations for usage of hermetic type of expanders and necessity for gear boxes. This will all add to maintenance cost and complexity of the system.

Depending on application, there will be an optimum power to achieve smaller sizes and cost for ORC. Square root of this optimum power is proportional to inverse of speed. As an example, for a 250 kW ORC, speed of 400 round per second would be the option. This can help to choose speed for other powers as well. Selecting right power and speed will cause good efficiencies and low cost for both mechanical and electrical equipment.

Using optimum speeds in ORC will omit the necessity to use vacuum and oil pumps, shaft seals and gear box which will make the system so cost effective.

It should be noted that working at not optimum speeds will necessitate usage of gear box. Gear boxes cannot be lubricated with low viscosity lubricants like working mediums, hence, optimum speed will help in removing the gear box and its lubrication consequently. Since removing oil can help the shaft seals to work properly; systems with no shaft outlets or hermetic design will be an appropriate option to achieve best component performance with usage of optimum speed.

In order to easily compensate the thrust load on thrust bearings via pressure distribution, and to have minimum starting torque on radial bearings, vertical shafts for rotors can be selected. The bearings can be lubricated with fluid in liquid or vapor states. This will help in more compact designs for ORC unit if right speed has also been chosen and no gear box is needed.

Applying the hermetic design will only needs a few labyrinth seals and the leakages will pass to condenser, while shaft outlets are removed and no tight sealing, lubrication or vacuum pumps are necessary. This makes the system so low maintenance. All in all, choosing the right speed for ORC will reduce the size of unit, complexity and cost of the system.

A study\(^\text{47}\) suggests that most efficient high speed ORC units are 250kW ones. Since they are modular units, they can be combined to produce more powers efficiently.

1-4 Turbine selection

Turbine or expander in ORC is normally the most costly part. This equipment can be up to about 60% of total cost of the system; as a result, selection of right turbine depending on application and the system specifications is an important task.

In this section, brief items about selection of different types of turbines or expanders are mentioned. (For more details on differences between turbines and expanders and their types refer to appendix, section A-2).

Based on preferred power range and ORC speed, degree of superheat or quality of inlet fluid of turbine, lubrication and sealing type, expander or turbine of ORC can be selected. In Figure 23 it is shown how the turbine can be chosen based on power range and rotational speed. In this figure, lines show that expanders have more operational speed with more powers. Even expanders with 10kW output and less than 5000 round per minute rotational speed are possible. In contrast for turbines, less rotational speed means more power. 

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In low power ranges, screw expanders with low rotational speeds and possibility for admission of wet fluid can be used. This can be compressors that are rotated in other direction. They can be lubricated by their own fluids. One type of screw expanders is one which has a male and female part. Male part has normally 4 lobes and female has 6 lobes. During rotation, trapped volume between these two sides will be expanded and can produce shaft work. Lysholm turbine which is a helical screw expander is one type of these expanders. Due to low speed of rotation, hermetic type of expanders can be coupled to generators easily without need for gearbox.

---

A new turbine type called Euler turbine can also be added to the list, especially for medium power ranges, e.g.: 250 kW. This turbine has less speed than radial inflow turbines of about half of that and can admit gasses with moisture, due to its centrifugal force which force the moisture out of turbine and since the turbine is to some extent erosion resistant. This turbine is radial out flow type and can be used without gear box. 

In rather higher power ranges radial inflow turbines can be chosen. They have several advantageous over axial turbines. They keep high efficiencies in small sizes and they are able to work in high pressure ratios. Since they have variable inlet nozzles, they can be well operated in variable load situations same as geothermal plants. Dry gas seal with ability of recovering leakages can also be added to the system. This type of turbine cannot admit wet fluid due to significant decrease in turbine efficiency and erosion problems in turbine.

For gaining higher powers in MW scales, axial turbines can also be selected; however, in ORC applications of less than 1 MW they are not efficient enough as axial turbines are designed for higher powers applications. 

1-5 Turbine size factor

Turbine size factor is defined as Equation 5.

Equation 5: Turbine size factor \( (SF_{\text{Turbine}}) = \frac{\nu^{0.5}}{\mu^{0.25}} \) (m)

In Equation 1, while \( \nu \) stands for volumetric flow in turbines and its unit is: \( m^3/s \), \( \mu \) is the isentropic enthalpy difference throughout the turbine in (KJ/Kg). Turbine size factor is shown by \( SF_{\text{Turbine}} \) here and the unit is meter. This can be used to compare different turbines sizes and is a proper indicator of its relative cost. More size factor means bulkier and more expensive turbines.

Pinch point temperature is the temperature difference between heat source temperature and evaporator temperature. If heat source temperature is assumed fixed, more pinch point temperature will result in less evaporator temperature and then with same heat input more superheat is done in ORC. In Table 9, from EES model, it is shown that higher pinch point temperature which means less saturation temperature in evaporator and more super heat, will result in larger turbines which will increase the turbine cost.

<table>
<thead>
<tr>
<th>( T_{pp} )</th>
<th>( T_{total} )</th>
<th>( x_{loss1} )</th>
<th>( x_{loss2} )</th>
<th>( x_{loss3} )</th>
<th>( SF_{\text{Turbine}} )</th>
<th>Isentropic Flow</th>
<th>( \eta_{exe} )</th>
<th>( \delta_{T} )</th>
<th>( \Omega_{net} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.1334</td>
<td>41.08</td>
<td>9.361</td>
<td>7.228</td>
<td>0.1879</td>
<td>0.06552</td>
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<td>21.19</td>
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</tr>
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<td>7.222</td>
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<td>9.229</td>
<td>7.401</td>
<td>0.1937</td>
<td>0.06903</td>
<td>0.7791</td>
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</tr>
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<td>7.574</td>
<td>0.1997</td>
<td>0.07273</td>
<td>0.7769</td>
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<td>1193</td>
</tr>
<tr>
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<td>0.07662</td>
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<td>13.98</td>
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<td>7.923</td>
<td>0.2123</td>
<td>0.08073</td>
<td>0.7724</td>
<td>25.36</td>
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<td>0.08507</td>
<td>0.7701</td>
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<td>1149</td>
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<td>8.275</td>
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<td>0.7627</td>
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<td>1099</td>
</tr>
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<td>8.055</td>
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<td>0.105</td>
<td>0.7601</td>
<td>31.55</td>
<td>1082</td>
</tr>
</tbody>
</table>

Table 9: Turbine size factor variation with evaporator saturation temperature and degree of superheat \( (T_{pp} \) is pinch point temperature in evaporator and \( \delta_{T} \) is degree of superheat)
Another parameter which can be introduced to compare different setups influence on turbine is the “isentropic volume flow ratio”. This is the ratio between isentropic outgoing and incoming volumetric flows with unit of m³/s. This is shown by isentropic flow ratio here or simply (Isentropic.flow). Since this parameter shows the change in the volume of fluid, it indicates the amount of compressibility of the fluid in the Turbine.

In Table 10 shows that more pinch point temperature and degree of supper heat means more isentropic flow ratio and consequently more compressibility effect from turbine is sensed. Isentropic flow ratio is an important factor since it is related to turbine efficiency. Isentropic flow ratios of above 0.5 are not interested because in this case the Turbine efficiency will be less than 80%. This is mainly true for axial single stage turbines; according to G. Angelino, C. Invernizzi, E. Macchi, (1991).  

<table>
<thead>
<tr>
<th>Tpp</th>
<th>( \eta_{\text{Total}} )</th>
<th>exeloss1</th>
<th>exeloss2</th>
<th>exeloss3</th>
<th>SF(_{\text{Turbine}} )</th>
<th>Isentropic Flow</th>
<th>( \eta_{\text{exe}} )</th>
<th>( \Delta_T )</th>
<th>Wnet</th>
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</thead>
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</tr>
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<td>0.118</td>
<td>0.118</td>
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<td>0.118</td>
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</tbody>
</table>

Table 10: Isentropic flow ratio change with pinch point and degree of superheat modifications

---

1-6 Pump efficiency

Pump efficiency can influence the ORC performance to an important extent. More pump efficiency means that the pump is able to increase the fluid pressure before evaporator to higher degrees and less output power of ORC will be used for pump work. Higher pressure increase from pump means larger evaporator pressure that is connected to the pump and this is usually interesting (as already discussed). As a result it is important to choose a right pump for the system to gain desired amount of pressure after the pump.

Some special pumps same as very accurate double diaphragm pumps minimize the leakage. Hence, in case of flammable or toxic fluids they are used to minimize the leakage and corresponding risk. Centrifugal pumps are used for high capacities.

To scrutinize how the pump efficiency affects the ORC performance, different values for that is tested in the generated computer model. Equations 1, 6 and 7 are used to get the values for Pump work, and system net output work depending on pump efficiency. These data are needed to complete the cycle calculations. “m” stands for mass flow rate of fluid in cycle, \( h_2 \) is fluid enthalpy after pump and \( h_1 \) is fluid enthalpy before pump. \( \eta_{pump} \) is pump efficiency, \( W_{turbine} \) shows the gained power from turbine and \( W_{net} \) is total net work output.

**Equation 6: Pump work**

\[
\text{Pump work} = m \times (h_2 - h_1) \times \eta_{pump}
\]

**Equation 7**

\[
W_{net} = W_{turbine} - \text{Pump work}
\]

Then ORC thermal efficiency is achieved from equation 1. Table 11 from EES program shows that pump efficiency has influence on total ORC efficiency and output power. Better pump efficiency gives higher work output and ORC thermal efficiency.

<table>
<thead>
<tr>
<th>( \eta_{pump} )</th>
<th>( \eta_{ORC} )</th>
<th>( W_{net} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
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</tr>
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<td>0.8444</td>
<td>0.1548</td>
<td>874.4</td>
</tr>
<tr>
<td>0.9</td>
<td>0.1548</td>
<td>875.9</td>
</tr>
</tbody>
</table>

**Table 11: Effect of pump efficiency on evaporator pressure and total cycle efficiency**
1-7 ORC fluid mass flow

Each pump has its own characteristics and a specific relation between its efficiency and volume/mass flow rates. An example of pump efficiency relation with mass flow rate is shown in equation 8 and 9. "m" is the fluid mass flow rate (Kg/s) and is related to volume flow rate Q (m³/s) and density (inverse of specific volume at point 1 which denotes density at pump inlet; see equation 9).

Efficiency equation of pump which is a second order equation shows that, efficiency peaks at certain mass flow rates (The function has one peak value, see equation 8). As a result, if variation of fluid mass flow in ORC results in better pump efficiency, the overall cycle total efficiency will be improved. Table 12 is an example of this, since in the examined mass flow ranges, the chosen pump has better efficiency in higher mass flow rates (the efficiency function gets higher values with higher mass flow rates), ORC performance is improved with higher mass flow rates.

The restriction on highest favorable mass flow rate will be:

1- Pinch point temperature of evaporator - critical temperature of fluid: Evaporator pressure (which is related to pump mass flow rate and it pressure increase) and accordingly its temperature cannot increase above critical temperature of ORC working fluid. Moreover, it is known that pinch point temperature is heat source and evaporator temperature difference. If evaporator temperature increases a lot, pinch point temperature is decreased. This is in contrary with evaporator design; hence evaporator temperature and pressure cannot increase after certain values.

2- Maximum efficiency of pump: Pump efficiency will decrease with higher mass flow rates after it has reached its maximum value (This is obvious from pump efficiency curves derived from equation 8).

Equation 8:

\[ \eta_{\text{pump}} = 0.0714 + 17.973 \cdot Q - 0.8862 \cdot Q^2 \]

Equation 9:

\[ Q = m \cdot \text{specific.volume}_1 \]

![Table 12: Variation of ORC thermal efficiency with mass flow](image)
Normally in heat source side, there is a fluid which carries heat. This can be a high pressure water, or oil, etc. More heat source fluid mass flow (mw) means more available heat and this will increase the ORC efficiency and output power (According to table 13). This is also notable that this will increase the turbine size which will to some extent offset the reduced cost (According to Table 13). It should be noted that for each application of ORC, mass flow rate of the system shall be optimized based on that application. For example in case of solar collector and ORC system with single working fluid, mass flow rate shall be optimized to give best system efficiency not only ORC best efficiency. This includes the collector efficiency as well.

<table>
<thead>
<tr>
<th>mw</th>
<th>Tpp</th>
<th>( \eta_{\text{tot}} )</th>
<th>( \eta_{\text{exe}} )</th>
<th>( \delta_T )</th>
<th>Wnet</th>
<th>exloss1</th>
<th>exloss2</th>
<th>SF_turbine</th>
<th>exloss3</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>43.5</td>
<td>0.09238</td>
<td>0.6768</td>
<td>5.329E-14</td>
<td>563.5</td>
<td>78.59</td>
<td>4.102</td>
<td>0.2736</td>
<td>2.496</td>
</tr>
<tr>
<td>22.22</td>
<td>43.5</td>
<td>0.09649</td>
<td>0.6868</td>
<td>5.329E-14</td>
<td>647.2</td>
<td>75.37</td>
<td>4.69</td>
<td>0.2926</td>
<td>2.744</td>
</tr>
<tr>
<td>24.44</td>
<td>43.5</td>
<td>0.09603</td>
<td>0.6949</td>
<td>5.329E-14</td>
<td>730.3</td>
<td>72.14</td>
<td>5.278</td>
<td>0.3068</td>
<td>2.992</td>
</tr>
<tr>
<td>26.67</td>
<td>43.5</td>
<td>0.1002</td>
<td>0.704</td>
<td>5.329E-14</td>
<td>814.5</td>
<td>68.91</td>
<td>5.666</td>
<td>0.323</td>
<td>3.24</td>
</tr>
<tr>
<td>28.89</td>
<td>43.5</td>
<td>0.1010</td>
<td>0.7129</td>
<td>5.329E-14</td>
<td>897.2</td>
<td>65.69</td>
<td>6.272</td>
<td>0.3359</td>
<td>3.699</td>
</tr>
<tr>
<td>31.11</td>
<td>43.5</td>
<td>0.1019</td>
<td>0.7206</td>
<td>9.503</td>
<td>967.3</td>
<td>62.34</td>
<td>6.418</td>
<td>0.3462</td>
<td>4.396</td>
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<tr>
<td>33.33</td>
<td>43.5</td>
<td>0.1015</td>
<td>0.7276</td>
<td>24.3</td>
<td>1032</td>
<td>58.49</td>
<td>6.517</td>
<td>0.354</td>
<td>6.8</td>
</tr>
<tr>
<td>35.66</td>
<td>43.5</td>
<td>0.1009</td>
<td>0.7342</td>
<td>39.4</td>
<td>1094</td>
<td>54.11</td>
<td>6.59</td>
<td>0.3607</td>
<td>9.347</td>
</tr>
<tr>
<td>37.78</td>
<td>43.5</td>
<td>0.1</td>
<td>0.7406</td>
<td>54.59</td>
<td>1152</td>
<td>49.25</td>
<td>6.644</td>
<td>0.3666</td>
<td>12.48</td>
</tr>
<tr>
<td>40</td>
<td>43.5</td>
<td>0.09508</td>
<td>0.7467</td>
<td>69.75</td>
<td>1209</td>
<td>43.93</td>
<td>6.687</td>
<td>0.3718</td>
<td>15.14</td>
</tr>
</tbody>
</table>

Table 13: Heat source fluid mass flow variation effect on ORC efficiency and output power
1-8 Heat exchangers

Heat exchangers can use up to 30% of total ORC cost. They consist of evaporator, regenerator and condenser.\textsuperscript{22}

According to a study\textsuperscript{50} higher evaporator pressure results in less overall heat exchanger area in ORC. This includes the total heat transfer area in evaporator, condenser, de-supper heater (explained in appendix, section: A-2), and pre-heater. To increase evaporator pressure, pump should be adjusted in order to give higher pressures in evaporator inlet. Moreover, as already mentioned, reasonable higher evaporator pressure is interesting, since overall turbine size will be decreased and this will bring down the cost. However, excessive high pressures in heat exchangers result in more expensive materials for its manufacturing. Hence, a balance on chosen pressure should be gained. In some applications, it is recommended to keep this pressure bellow 20 bars to prohibit expensive materials for heat exchanger manufacturing (Ulli Drescher, Dieter Bru¨ggemann, 2007).

1-9 Boilers – evaporators

Boilers are normally divided into two sections which are both shell and tube heat exchangers. One is responsible for pre-heating the liquid working fluid to boiling temperature, while the other section will evaporate it. Normally exhaust gas is in the shell while the ORC working fluid moves in tube side. To avoid corrosion the tube part is from copper or stainless steel and shell part is cylindrical to increase the heat transfer area.\textsuperscript{22}

Vapor and liquid separation is done between boiler shell and casing in outside part. Since only dry vapor is admitted to turbine, a signal from a float control valve will be needed. This will adjust feed pump speed or mass flow recirculation from pump exit to pump inlet in order to control the flow into to boiler and receive dry vapor after boiler.

If vapor can be admitted to turbine, a single pass system will be used to give desired wetness of vapor and external recirculation will not be needed any more. In this case, a cheaper type of plate fin can be used. A vapor fraction control system however shall be added.

Normally this control system uses a reference data from fluid after turbine. A small fraction of flow from boiler outlet is throttled to condenser inlet. This sample must be supper heated when reaching condenser pressure. By knowing the sample pressure and temperature at condenser inlet its enthalpy is measured. By equating this

enthalpy with the enthalpy after turbine and knowing boiler discharge pressure, moisture content of medium after boiler will be calculated. A microprocessor is foreseen to do these simple control measures. Normal ORC boilers along with boilers of ORCs with wet admission ability in turbine are depicted in Figure 24.\textsuperscript{50}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure24.png}
\caption{A- Normal boiler system \hspace{1cm} B- Boiler system with possibility of wet vapor outlet \hspace{1cm} Source: ASME International Mechanical Engineering Congress and Exposition\textsuperscript{51}}
\end{figure}

\section*{1-10 Regenerator}

Main responsibility of this section is to improve the cycle efficiency by pre-heating the ORC working fluid exiting the pump. This is done by extracting heat from superheated fluid after the expander.

For regenerator usage several issues shall be considered. First of all, if ORC fluid is dry then using regenerator is reasonable. Secondly, it should be noted that for applications where final exhaust temperature in heat source side is fixed using regenerator is considerable. Imagine that exhaust medium of heat source will be used in district heating system and needs to be in a certain temperature. In this case regenerator can be considered. Otherwise, regenerator will yield in more evaporator inlet temperature for ORC fluid and this will results in less heat recovery from heat source.

This is true that using regenerator will cause less irreversibility in evaporator and gives better heat transfer, but on the whole, with variable heat source, less heat is extracted from heat source since higher temperature ORC

\textsuperscript{51} H. Leibowitz, I. K. Smith, N Stosic, Cost effective small scale ORC systems for power recovery from low grade heat sources, ASME International Mechanical Engineering Congress and Exposition 2006; Pages 5-6.
fluid has been entered in evaporator and can absorb less heat. This is important especially in ORC as cooling system or where more heat recovery rather than fixed heat source is interesting. Even if regenerator gives better efficiency, as mentioned, it might not extract heat in best way and will not give higher power generation always. Hence, using this costly equipment shall be well justified to compensate its price.

It is notable that unlike regenerator and evaporator, in condenser superheated ORC working fluid passes through shell and the cooling fluid is in tube side while shell and tube heat exchangers are used.22

1-11 Working fluid

Regarding the working fluid several criteria as here bellow shall be considered:

- It should cause high efficiency and power output for ORC
- Not so much pump work
- High stability and low flammability
- Low cost and being available
- Low global warming and ozone depletion potential
- Low toxicity
- Low corrosion
- Rather high boiling temperature and melting point
- Rather high latent heat but not high specific heat, first one to increase heat recovery, latter one to decrease the cost of condenser due to decrease of load.
- High thermal conductivity and better match with heat source temperature to have better heat transfer
- Wet and isentropic fluids will have most area under T-S diagram, however, dry fluids have less vapor formation during expansion and by using them internal heat exchanger will be more useful for efficiency improvement. Wet fluids shall be used normally with high superheats to prevent liquid formation which is not interesting (Wet, isentropic and dry fluids are explained in appendix, section A-2).
- High critical temperature depending on working temperatures. ⁵² ⁵³

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2- Optimization of Micro ORC as CHP or CCHP system mainly based on solar energy

Using the optimized ORC system will provide low cost heat, electricity and cooling while CO₂ and other emissions are minimized in the system and fuel consumption is reduced. This ORC can be combined to a CHP or CCHP in order to use its waste heat and produce more power or can be the standalone power producer of the system.

The system of combined solar collector and ORC which is a CHP system can be optimized for best power output by choosing proper collector water and ORC fluid mass flow rates, ORC components properties and fluid type.

In chapter 2 biomass based CHP and ORC system with rather high temperatures and high produced power is studied. In this section it is aimed to investigate criteria of ORC in smaller scale and lower temperature applications. To gain this, the generated computer model is modified for lower grade heat source. Alternative fluids, different condenser and evaporator temperatures, etc has been checked and some results are introduced hereafter.

2-1 System details and alternatives for ORC - CHP system

The system under study is a solar based ORC that can be connected to a heat pump which can be used for cooling or heating purposes. As known, the produced power can be applied for electricity generation, irrigation especially in remote areas, pumping, water desalination or cooling, etc. A typical system is shown in Figure 9. Additional burner can be added to increase availability of system. In Figure 24 another small CHP system using ORC as additional power generation system is depicted to illustrate alternatives for micro ORC usage in CHP plants especially in commercial and small scales. Power generation unit (PGU) is completed by ORC. Heating system, hot water section and vapor compression system (VCS) is also presented in the Figure 25.

Depicted system in Figure 9 is a small CHP system which uses ORC as only power producer while for the system in Figure 25 main power producer (PGU) is completed by ORC to recover waste heat and produce more electricity. Both systems are in small scales and have commercial usages to provide cooling, heat and electricity for buildings.
To clarify figure 25, the system poses a power generation unit (PGU). Waste heat from this unit can be either used in ORC or directly heating system of buildings. Heating in buildings are in forms of Hot water or heating coil. Electricity from ORC and PGU will be fed to grid.

2-2 Working fluids

For solar applications, different setups of model and available results from a study by Joan Carles Bruno, Jesu’s Lo´pez-Villada, Eduardo Letelier, Silvia Romera, and Alberto Coronas, (2008) shows that Benzene series same as n-propilbenzene will give better efficiencies for ORC; however cost and safety issues shall be considered as well. In general, isentropic fluids and especially wet fluids give more working area in T-S diagram and better efficiencies rather than dry fluids. However, wet expansion of wet fluids limit their use, so isentropic and dry fluids are more applied (T.C. Hung a, S.K. Wang a, C.H. Kuo b, B.S. Pei c, K.F. Tsai, 2010).

---

2-3 Heat transfer area
For solar based ORC it is two possibilities to transfer heat into ORC. One method is to use ORC fluid in direct connection with collector, and the other is to use another fluid to transfer heat from collector to ORC fluid. In first case total needed heat transfer area is about 20m²/kWh and for second case it is about 25m²/kWh.\textsuperscript{55}

2-4 Optimized temperature based on Collector and ORC efficiencies
System under study is fairly similar to system in Chapter I-3. Here, an intermediate fluid can works between ORC evaporator and solar collector (In some applications this fluid is ORC fluid itself). The collector heat is transferred to ORC boiler (evaporator) by the intermediate fluid and power is generated in ORC. Even waste heat of system can be used in a heat pump to meet the thermal load. Since in this study, the system is investigated for its ORC performance, following electricity load is much interesting option rather than focusing on thermal load. However, ORC can only be in operation while available recovered thermal energy is more than thermal load of system.

\textbf{Figure 26: Solar ORC}

\textsuperscript{55} Delgado-Torres AM, García-Rodríguez L. Analysis and optimization of the low-temperature solar organic Rankine cycle (ORC). Energy Convers Manage (2010), doi:10.1016/j.enconman.2010.06.022
It is known that by increasing maximum working temperature, ORC efficiency will increase. This is while that solar collector efficiency will decrease by increasing the temperature. As a result, in solar based ORC with a collector system, a tradeoff between best collector and ORC efficiencies shall be found.

In the generated computer model, ORC thermal efficiency has been calculated. In order to add solar collector efficiency ($\eta_{c}$), Equation 12 is used. Constants of this equation depend on collector type. As a case in point, for a compact parabolic through collector constants are: eff0: 0.75, a: 0.11232 and b: 0.00128 (Bruno, 2008). In equation 12, $T_{int}$ is collector temperature (equal to $T_{water1}$ in figure 26), $T_0$ is ambient temperature, and $I$ which is solar global radiation - assumed to be 1000 (W/m2).

**Equation 10:** $\eta_{c} = eff_0 - a \left( \frac{T_{int}-T_0}{I} \right) - b \cdot I \cdot \left( \frac{T_{int}-T_0}{I} \right)^2$

In table 14, values for other type of collectors are given as well.

<table>
<thead>
<tr>
<th>Collector Type</th>
<th>Flat plate collector</th>
<th>Evacuated tube collector</th>
<th>Compact parabolic trough collector</th>
</tr>
</thead>
<tbody>
<tr>
<td>eff0</td>
<td>0.768</td>
<td>0.665</td>
<td>0.75</td>
</tr>
<tr>
<td>a</td>
<td>2.9</td>
<td>0.59</td>
<td>0.11232</td>
</tr>
<tr>
<td>b</td>
<td>0.0108</td>
<td>0.0019</td>
<td>0.00128</td>
</tr>
<tr>
<td>Collector area</td>
<td>2.307</td>
<td>1.088</td>
<td>2.00</td>
</tr>
</tbody>
</table>

**Table 14: Solar Collector data**


More data for other collectors can be found in literature same as a study by Joan Carles Bruno, Jesu´s Lo´pez-Villada, Eduardo Letelier, Silvia Romera and Alberto Coronas , (2008).

In Table 15, effect of intermediate fluid temperature at collector exit ($T_{water1}$ as shown in figure 26) on system performance is depicted. As known, more evaporator inlet temperatures will yield better ORC efficiencies. This is while that more collector temperature will decrease the collector efficiency due to increase of thermal losses. On the whole, overall system efficiency which is from multiplication of collector and ORC efficiencies can be optimized for best intermediate fluid temperature and collector temperature as in Figure 27.

$T_{water_3}$ (See figure 26) is defined as intermediate fluid temperature at outlet of ORC evaporator (boiler); and here it is assumed to be about 90°C. This fluid with this or higher temperatures can be used in absorption chillers (Heat pump) or district heating system. The other option is to use intermediate fluid before entrance to ORC evaporator in a district heating system.
Table 15: Effect of intermediate fluid temperature on collector, ORC and total system performance

<table>
<thead>
<tr>
<th>T_{water1}</th>
<th>\eta_{ORC}</th>
<th>\delta_T</th>
<th>\eta_{T_{collect}}</th>
<th>\eta_{system}</th>
<th>W_{net}</th>
<th>Q_{1} [kW]</th>
<th>\eta_{T_{steam}}</th>
<th>S_{Turbine}</th>
<th>P_{1} [kPa]</th>
<th>T_{sat}</th>
<th>CF</th>
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<th>\text{exeloss2}</th>
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<td>0.7202</td>
<td>0.08974</td>
<td>470.8</td>
<td>3778</td>
<td>0.834</td>
<td>0.337</td>
<td>576.1</td>
<td>84.39</td>
<td>0.3408</td>
<td>2.418</td>
<td>6.074</td>
<td>4.684</td>
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</tr>
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<td>136.7</td>
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<td>11.9</td>
<td>0.7195</td>
<td>0.09022</td>
<td>491.2</td>
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<td>0.3411</td>
<td>581.2</td>
<td>84.76</td>
<td>0.3434</td>
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<td>4.741</td>
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<tr>
<td>138.3</td>
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<td>13.2</td>
<td>0.7188</td>
<td>0.09071</td>
<td>511.9</td>
<td>4056</td>
<td>0.8077</td>
<td>0.345</td>
<td>586.2</td>
<td>85.14</td>
<td>0.346</td>
<td>5.094</td>
<td>6.147</td>
<td>4.8</td>
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<td>0.7553</td>
<td>0.3437</td>
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<td>15.78</td>
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<td>0.09167</td>
<td>553.9</td>
<td>4334</td>
<td>0.7833</td>
<td>0.3523</td>
<td>596.6</td>
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<td>10.63</td>
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<td>19.64</td>
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<td>0.09311</td>
<td>618.5</td>
<td>4751</td>
<td>0.7497</td>
<td>0.3622</td>
<td>612.5</td>
<td>87.03</td>
<td>0.3593</td>
<td>12.05</td>
<td>6.332</td>
<td>5.142</td>
<td>0.194</td>
</tr>
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<td>148.3</td>
<td>0.131</td>
<td>20.92</td>
<td>0.7145</td>
<td>0.09358</td>
<td>640.5</td>
<td>4890</td>
<td>0.7393</td>
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<td>618</td>
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<td>0.3648</td>
<td>14.93</td>
<td>6.407</td>
<td>5.299</td>
<td>0.1909</td>
</tr>
</tbody>
</table>

Figure 27: Effect of collector temperature on system performance
2-5 Effect of degree of superheat

Another setup of the program showed that if degree of superheat in evaporator is kept constant while more heat is given to the system it will be more efficient. This effect is shown in Table 16. To keep the degree of superheat constant, more mass flow rate in ORC is used and this will increase the pump work and evaporator pressure.

![Table 16: Effect of constant superheat on turbine size factor (SF) and Heat exchangers factor (HEF)](image)

2-6 Condenser temperature and Pressure

Since using waste heat of ORC from condenser necessitates more than normal temperatures in condenser, effect of condenser temperature and pressure on ORC performance is checked by the computer model.

As in Table 17 is shown, having more condenser temperatures (Tconwat2 as shown in figure 25) means less ORC efficiency and output power. On the other hand, more condenser temperature will give more potential for connection of a heat pump to an ORC condenser and to produce additional heat. As a result, it is an important issue to check if added heat capacity can compensate the loss in electricity power when local electricity and heat prices are concerned.
Table 17: Effect of condenser temperature on ORC efficiency and output power
3- Optimization of ORC for cooling purposes

ORC can be used for cooling purposes. One example is cooling of electrical devices during operation.

As in Figure 28 is shown, an intermediate fluid takes the heat from the device (=Technical equipment) which needs to be cooled to ORC unit. Then, ORC evaporator (5-1) and pre-heater (4-5) will gain the heat from cooling circuit. ORC produces power from the gained heat from the device and the device is cooled. If needed, additional cooling by extra source in 03-04 is introduced to gain lower temperature in point 04.

![Figure 28: ORC for cooling devices](https://via.placeholder.com/150)

By assuming fixed amount of cooling of device by ORC evaporator(5-1), less enthalpy change in evaporator requires more mass flow rate (m) of fluid to extract the same amount of heat and cool the device in evaporator(\(Q_{\text{evaporator}}=m\Delta h_{\text{evaporator}}=\text{cte}\)). To achieve higher mass flow rates, pump will be adjusted to increase the mass flow rate. On the other hand, more enthalpy change in pre-heater means better heat extraction from heat source in pre-heater (4-5), since (\(Q_{\text{pre-heater}}=m\Delta h_{\text{preheater}}\)). More mass flow rate and more gained heat in ORC means more power output from ORC unit (\(W=m\Delta h_{\text{turbine}}\)). Due to given reasons, if enthalpy change in evaporator is becomes less and that in pre-heater becomes higher, better cooling, less need for additional cooling device and more power from ORC are expected (W. Nowak, A. B-Gozdur, A.A. Stachel, 2008).

To investigate effect of different factors on cycle, a number called cooling factor (CF) is defined as equation 13. This factor is the ratio of enthalpy change in pre-heater (4-5) to enthalpy change in evaporator (5-1). As mentioned, higher enthalpy change in pre-heater and lower enthalpy change in evaporator is interesting and this means higher CF number.

---

Equation 11: \[ CF = \frac{\Delta h_{\text{pre-heater}}}{\Delta h_{\text{evaporator}}} \]

In Figure 29 evaporation process is shown by red line while pre-heating is in green color.

In Table 18 effect of evaporator pressure on CF is depicted. It shows that more evaporator pressure means more cooling factor which is more interesting. As in Table 18 is obvious, more CF means lower \( T_{\text{water2}} \) (\( T_{\text{water2}} \) stands for intermediate fluid exhaust temperature at evaporator outlet at point \( \theta2 \)). The lower this temperature, it means more cooling has been occurred. One restriction for evaporator pressure increase is pinch pint temperature (\( T_{\text{pp}} \)) which shall be kept above zero (Refer to section II, 1).
Table 18: Effect of evaporator pressure on CF

Some different working fluids for ORC, i.e.: R227ea, R134a and R254fa (with acceptable GWP) have been tasted to compare their CF, ORC efficiencies and performance here.

In Table 19, constant heat source in evaporator has been assumed ($Q_{\text{Evaporator}}$ in KW assumed to be constant). As table shows, more mass flow of ORC fluid means more ORC power ($W=m\cdot\Delta h_{\text{turbine}}$), better CF and better performance from ORC.

In another setup of program, amount of heat generation in Technical equipment (device) is assumed constant. Then by running the model for different fluids, it is concluded that in comparison with R134a and R245fa; R227ea can cause more cooling in the device and produce more power in ORC. This is shown in Table 20 as
Twater3 (exhaust intermediate fluid temperature from pre-heater at point 03) for R227ea is less than values in Table 21 for R134a, Table 22 for R254fa. Second and third ranks go for R134a and R245fa.

It is clear that R227ea and R134a are among those fluids which can extract more heat in pre-heater rather than evaporator since they can achieve CF of of more than 1. For R134a this result is in consistent with a previous study by Tchanche.

<table>
<thead>
<tr>
<th>$P_1$ [kPa]</th>
<th>Twater3</th>
<th>CF</th>
<th>Tpp</th>
<th>$\eta_{total}$</th>
<th>$\eta_{exe}$</th>
<th>SF_Turbine</th>
<th>Wnet</th>
</tr>
</thead>
<tbody>
<tr>
<td>1838</td>
<td>52.88</td>
<td>1.021</td>
<td>10.6</td>
<td>0.1601</td>
<td>0.6132</td>
<td>0.2823</td>
<td>677.7</td>
</tr>
<tr>
<td>1880</td>
<td>51.31</td>
<td>1.064</td>
<td>9.584</td>
<td>0.1655</td>
<td>0.6161</td>
<td>0.2788</td>
<td>700.5</td>
</tr>
<tr>
<td>1919</td>
<td>49.76</td>
<td>1.106</td>
<td>8.637</td>
<td>0.1670</td>
<td>0.6187</td>
<td>0.2757</td>
<td>722.6</td>
</tr>
<tr>
<td>1956</td>
<td>48.23</td>
<td>1.148</td>
<td>7.754</td>
<td>0.1658</td>
<td>0.6211</td>
<td>0.2724</td>
<td>744.1</td>
</tr>
<tr>
<td>2091</td>
<td>46.71</td>
<td>1.19</td>
<td>6.928</td>
<td>0.1807</td>
<td>0.6232</td>
<td>0.2706</td>
<td>765.0</td>
</tr>
<tr>
<td>2024</td>
<td>45.2</td>
<td>1.231</td>
<td>6.155</td>
<td>0.1856</td>
<td>0.6252</td>
<td>0.2685</td>
<td>785.4</td>
</tr>
<tr>
<td>2056</td>
<td>43.71</td>
<td>1.272</td>
<td>5.428</td>
<td>0.1903</td>
<td>0.6269</td>
<td>0.2665</td>
<td>805.3</td>
</tr>
<tr>
<td>2086</td>
<td>42.23</td>
<td>1.313</td>
<td>4.745</td>
<td>0.1949</td>
<td>0.6285</td>
<td>0.2644</td>
<td>824.8</td>
</tr>
<tr>
<td>2115</td>
<td>40.76</td>
<td>1.353</td>
<td>4.102</td>
<td>0.1994</td>
<td>0.6309</td>
<td>0.2623</td>
<td>843.9</td>
</tr>
<tr>
<td>2142</td>
<td>39.3</td>
<td>1.393</td>
<td>3.495</td>
<td>0.2038</td>
<td>0.6321</td>
<td>0.2608</td>
<td>862.7</td>
</tr>
</tbody>
</table>

Table 20: R227ea ORC performance for cooling

<table>
<thead>
<tr>
<th>$P_1$ [kPa]</th>
<th>Twater3</th>
<th>CF</th>
<th>Tpp</th>
<th>$\eta_{total}$</th>
<th>$\eta_{exe}$</th>
<th>SF_Turbine</th>
<th>Wnet</th>
</tr>
</thead>
<tbody>
<tr>
<td>2273</td>
<td>67.01</td>
<td>0.6327</td>
<td>16.82</td>
<td>0.1284</td>
<td>0.711</td>
<td>0.2279</td>
<td>543.3</td>
</tr>
<tr>
<td>2444</td>
<td>63.89</td>
<td>0.7184</td>
<td>13.51</td>
<td>0.1405</td>
<td>0.7227</td>
<td>0.2202</td>
<td>594.7</td>
</tr>
<tr>
<td>2595</td>
<td>60.84</td>
<td>0.802</td>
<td>10.73</td>
<td>0.1517</td>
<td>0.7316</td>
<td>0.2145</td>
<td>642.5</td>
</tr>
<tr>
<td>2729</td>
<td>57.86</td>
<td>0.8838</td>
<td>8.358</td>
<td>0.1621</td>
<td>0.7386</td>
<td>0.2102</td>
<td>686.2</td>
</tr>
<tr>
<td>2848</td>
<td>54.93</td>
<td>0.9642</td>
<td>6.33</td>
<td>0.1719</td>
<td>0.7441</td>
<td>0.2073</td>
<td>727.7</td>
</tr>
<tr>
<td>2954</td>
<td>52.05</td>
<td>1.043</td>
<td>4.582</td>
<td>0.1812</td>
<td>0.7483</td>
<td>0.2046</td>
<td>767.0</td>
</tr>
<tr>
<td>3048</td>
<td>49.21</td>
<td>1.121</td>
<td>3.065</td>
<td>0.1901</td>
<td>0.7516</td>
<td>0.2029</td>
<td>804.5</td>
</tr>
<tr>
<td>3133</td>
<td>46.41</td>
<td>1.198</td>
<td>1.742</td>
<td>0.1986</td>
<td>0.7541</td>
<td>0.2017</td>
<td>840.5</td>
</tr>
</tbody>
</table>

Table 21: R134a ORC performance for cooling

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CONCLUSIONS

Based on performed investigation and market evaluation, ORC can be applied in several areas cost effectively. Biomass CHP and biomass digestion waste heat recovery applications, micro scale CHP system including solar CHP unit, solar desalination unit, ORC as micro gas turbine bottoming cycle to improve its efficiency from 30% to 40% and geothermal applications are some areas. In addition to that, waste heat recovery from steel, ceramic, and cement industries, IC engines, light and heavy duty vehicles same as ships can also be added to the list.

ORC is also able to be used as bottoming cycle of recuperative or high pressure ratio gas turbines and MOFCs cost effectively in comparison with other available technologies.

Solar based ORC seems to be implemented more cost effectively than PV power generation system with ability of energy storage in PCM storage. Even ORC can be used in gas cooled nuclear reactors to improve both thermal and exergy efficiencies. ORC can be used for cooling devices. In this application, optimization is done based on a number which is ratio of heat addition in pre-heater to heat addition in evaporator. The bigger the number better ORC performance is achieved.

To improve ORC system and to reduce the cost, trend of thermal and exergy efficiencies and turbine size showed meaningful relations with degree of superheat, mass flow rates of fluids, working fluid and expander or turbine types. Some of results were investigated to match with available information in literature.

Proper ORC speed will make the system to be compact and simple. This will eliminate the need for gear box and lubrication system while the pump, turbine and generator can be on a single shaft. Putting ORC components in vertical position compensates the thrust load on thrust bearings via pressure distribution, and causes minimum starting torque on radial bearings.

![Table 212: R245fa ORC performance for cooling](image)
It was noted that regenerator usage is interesting in dry fluids and where fixed heat source is introduced. However, even in these cases, better efficiency may not compensate the cost of regenerator.

More evaporator pressure gives better efficiencies. Pinch point temperatures, heat exchangers cost, critical temperature of working fluid would be a restriction for maximum working pressure of cycle. Consequently, optimum working pressure for cycle should be chosen.

ORC as in CHP or Trigeneration units are options to improve total efficiency and reduce the cost. For each ORC application, power and temperature range, best fluid can be chosen. For small scale solar CHP, Benzene series gives best results.

Optimization of ORC for cooling purposes showed that among five different fluids, R227ea give most cooling and ORC power. More working pressure was interesting while pinch point temperature is well kept above zero.

**Future work**

Each discussed application of ORC can be optimized based on same procedure which has been used in this project and the result can have a great impact on performance improvement and cost reduction of that specific application. This will give great potentials for a sophisticated power generations method with minimal environmental negative impacts.

There are opportunities to investigate each of aforementioned ORC applications in much more detailed and applied manner in further research activities. To name a few, heat recovery from slag in steel industry or internal combustion engines, Nano-structured metal-organic working fluids, geothermal or solar based cost effective ORC plants, efficient biomass based ORCs, micro ORCs etc, are candidates for more research activities.

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A- APPENDIX

A-1 Computer model

The generated computer model in EES is as follows:
Function exe (h, s, W)
    To := 20  # dead state temperature, Degree C
    ToK := 20 + 273.1
    xo := 1
    ho := h ( 'R11' , T=To , x=xo )
    so := s ( 'R11' , T=To , x=xo )  # specific entropy at dead state conditions
    h := enthalpy(R245fa, T=T, P=P)
    s := entropy(R245fa, T=T, P=P)
    exe := h - ho - ToK * ( s - so ) + W
End exe

Evaporator

Q₁ = m * \left[ \frac{h₂ - h₂_{satvap}}{evaeff1} + \frac{h₂_{satvap} - h₂_{satliq}}{evaeff2} \right]

evaeff1 = 0.9
evaeff2 = 0.8
m * ( h₂_{satliq} - h₁ ) = mw * ( hwₐ - hw₃ )
T_{water₃} = T ( 'Water' , P=P_{water} , h= hw₃ )
Q₁=8250
Q₁ = mw * ( hw₁ - hw₀ )
m=20
s₁ = s ( 'R11' , T=T₂ , x=0 )
T_{water₃} = 90
T_{water₁} = 150
Turbine
\[ \eta_{iS} = \frac{h_2 - h_3}{h_2 - h_{3s}} \]
\[ h_2 = h_{2satvap} \]
\[ \eta_{iS} = 0.8 \]
\[ W_{turb} = m \cdot (h_2 - h_3) \]
\[ S_3 = s \left( 'R11', h=13, P=P_2 \right) \]
\[ h_{2satliq} = h \left( 'R11', P=P_1, x=0 \right) \]
\[ h_{2satvap} = h \left( 'R11', P=P_1, x=1 \right) \]
\[ P_1=\text{Pressure}(R236fa, h=h_{2satliq}, x=0) \]

Cooling factor
\[ CF = \frac{h_{2satliq} - h_1}{h_{2satvap} - h_{2satliq}} \]

Condenser
\[ Q_2 = m \cdot (h_3 - h_4) \]
\[ Q_2 = mcw \cdot (h_{cw0} - h_{cw1}) \]

Pump
\[ \text{specvol}_4 = V \left( 'R11', T=T_2, x=0 \right) \]
\[ \text{specvol}_1=\text{Volume}(R123, h=1, \mu=P_1) \]
\[ \text{pumpwork} \cdot \frac{\eta_{pump}}{m} = (P_1 - P_2) \cdot \text{specvol}_4 \]
\[ \eta_{pump} = 0.7 \]
\[ \text{pumpwork}/\text{eta}_p = m \cdot (h_1 - h_4) \]
\[ h_1 = h_4 \]

Twater1=150
\[ T_{sup} = T_{water1} - \text{TPP1} \]
h_2 = h ('R11', P=P_1, T=T_{1sup})

T_{water} = 140

H_{pump} = 30 + 0.513*Q - 0.0964*Q^2
pumpwork = ((8.8*H_{pump})*(specificvol_4*1000))/Q/\eta_pump

\eta_pump = 0.714 + 17.973*Q - 0.8862*Q^2
pumpwork = ((8.8*H_{pump})*(specificvol_4*1000))/Q/\eta_pump

Q = m*specificvol_4

h_4 = h_1

s_4 = Entropy(R245fa, T=T_{2s}, P=P_1)

h_{1gs} = Enthalpy(R245fa, s=s_4, P=P_1)

\eta_pump = (h_{1gs} - h_4)/(h_1 - h_4)
\eta_pump = 0.7

Regenerator

(h_{sv} - h_1) = \eta_{reg}*(h_3 - h_{co})
\eta_{reg} = 0.9

Enthalpies

h_4 = h ('R11', T=T_{2s}, x=0)

T_{1sup} = T_{1sat}

T_{1sat} = T_{water} - T_{pp}

T_{1sup} = Temperature(R236fa, P=P_1, h=h_2)

h_2 = Enthalpy(R236fa, P=P_1, T=T_{1sup})

h_2 = Enthalpy(R245fa, T=T_{1sup}, x=P_1)

s_2 = s ('R11', T=T_{1sup}, P=P_1)

h_{3gs} = h ('R11', P=P_2, s=s_2)

h_4 = Enthalpy(R245fa, T=T_{2s}, x=0)

Conditions

T_{1sat} = 110

T_{1sup} = T_{1sat} + \delta T
\[ T_2 = T_{ppc} + T_{conwat2} \]

\[ T_{ppc} = 5 \]

\[ P_1 = \text{Pressure(R245fa, } T=T_{1sat}, x=1) \]

\[ P_2 = P(‘R11’, T=T_2, x=0) \]

\[ T_{1sat} = T(‘R11’, P=P_1, x=0) \]

\[ T_{pp2} = T_{water2} - T_{1sat} \]

\[ T_{pp2} = 15 \]

**Efficiency**

\[ \eta_{ORC} = \frac{W_{turb} - \text{pumpwork}}{Q_1} \]

\[ W_{net} = W_{turb} - \text{pumpwork} \]

**Exergy efficiency**

\[ e_{ex \text{ tot.\ loss}} = e_{\text{ex loss1}} + e_{\text{ex loss2}} + e_{\text{ex loss3}} \]

\[ e_{ex \text{ tot.\ in}} = e_{\text{ex in1}} + e_{\text{ex out1}} + e_{\text{ex out2}} + e_{\text{ex wat in}} + e_{\text{ex con wat in}} \]

\[ \eta_{ex} = 1 - \frac{e_{ex \text{ tot.\ loss}}}{e_{ex \text{ tot.\ in}}} \]

**Evaporator**

\[ e_{ex \text{ in1}} = \text{exe}(h_1, s_1, W_1) \]

\[ W_1 = 0 \]

\[ e_{ex \text{ out1}} = \text{exe}(h_2, s_2, W_1) \]

**Water side**

\[ m_w = 25 \]

\[ P_{\text{water}} = 500 \text{ Flue gas side} \]
Flue gas side

\[ T_{\text{water2}} = \text{Temperature}(\text{water}, P = P_{\text{water}}, h = h_{\text{w2}}) \]

\[ x_{\text{w1}} = x(\text{Water}, T = T_{\text{water1}}, P = P_{\text{water}}) \]

\[ x_{\text{w2}} = x(\text{Water}, T = T_{\text{water1}}, P = P_{\text{water}}) \]

\[ h_{\text{w1}} = h(\text{Water}, T = T_{\text{water1}}, P = P_{\text{water}}) \]

\[ s_{\text{w1}} = s(\text{Water}, T = T_{\text{water1}}, P = P_{\text{water}}) \]

\[ h_{\text{w0}} = h(\text{Water}, T = T_{\text{water2}}, P = P_{\text{water}}) \]

\[ s_{\text{w0}} = s(\text{Water}, T = T_{\text{water2}}, P = P_{\text{water}}) \]

\[ \text{exewat}_{\text{in}} = \text{exe}(h_{\text{w1}}, s_{\text{w1}}, W1) \]

\[ \text{exewat}_{\text{out}} = \text{exe}(h_{\text{w0}}, s_{\text{w0}}, W1) \]

\[ \text{exeloss1} = \text{exewat}_{\text{in}} - \text{exewat}_{\text{out}} - (\text{exe}_{\text{out1}} - \text{exe}_{\text{in1}}) \]

Turbine

\[ \text{exe}_{\text{out2}} = \text{exe}(h_3, s_3, W1) \]

\[ \text{exeloss2} = \text{exe}_{\text{out1}} - \left[ \text{exe}_{\text{out2}} + \frac{W_{\text{turb}}}{m} \right] \]

Condenser

\[ \text{exe}_{\text{out3}} = \text{exe}_{\text{in1}} \]

\[ T_{\text{conwat1}} = 7 \quad \text{Heat sink temperature} \]

\[ T_{\text{conwat2}} = 40 \]

Heat sink temperature

\[ P_{\text{conwat}} = 100 \quad \text{Heat sink side} \]

\[ h_{\text{cw1}} = h(\text{Water}, T = T_{\text{conwat1}}, P = P_{\text{conwat}}) \]

\[ s_{\text{cw1}} = s(\text{Water}, T = T_{\text{conwat1}}, P = P_{\text{conwat}}) \]

\[ h_{\text{cw0}} = h(\text{Water}, T = T_{\text{conwat2}}, P = P_{\text{conwat}}) \]
\[ \text{scw}_0 = s \left( \text{Water}, T=T_{\text{conwat2}}, P=P_{\text{conwat}} \right) \]
\[ \text{exe.conwat}_n = \text{exe} \left( h_{cw1}, \text{scw}_1, W1 \right) \]
\[ \text{exe.conwat}_\text{out} = \text{exe} \left( h_{cw0}, \text{scw}_0, W1 \right) \]
\[ \text{exeloss}_3 = \text{exe}_{\text{out2}} - \text{exe}_{\text{out3}} + \text{exe.conwat}_n - \text{exe.conwat}_\text{out} \]

**Turbine size factor**

\[ v_2 = v \left( R11', T=T_{12}, p_1=p_2, h=h_2 \right) \]
\[ \dot{V} = m \cdot v_2 \]
\[ H = h_2 - h_{36} \]
\[ \text{SF}_{\text{Turbine}} = \frac{\sqrt{v_2}}{H^{0.25}} \]

**Compressibility factor**

\[ v_{36} = v \left( R11', T=T_{12}, P=P_2, h=h_{36} \right) \]
\[ \text{Isentropic Flow ratio} = \frac{v_2}{v_{36}} \]

**Heat exchanger factor**

\[ l_{\text{total}} = m \cdot \text{ToK} \cdot \left[ \frac{h_2 - h_1}{T_{\text{water1}} + 273} - \frac{h_{36} - h_4}{\text{ToK} + 273} \right] \]
\[ \text{HEF} = \frac{Q_1 - l_{\text{total}}}{Q_1} \]
\[ \text{ToK} = 25 + 273 \]

**Solar collector**

\[ \text{tte}_0 = \text{tte}_0 - a \cdot \left[ \frac{T_{\text{water1}} - \text{Tamb}}{lr} \right] - b \cdot \left[ \frac{T_{\text{water1}} - \text{Tamb}}{lr} \right]^2 \]
\[ \text{tte}_0 = 0.75 \]
\[ a = 0.11232 \]
\[ b = 0.00128 \]
Ir = 1000
Tamb = 20

\eta_{system} = \eta_0 \cdot \eta_{ORC}

Cycle Points
Tpoint_1 = T ( 'R11' , P = P_1 , h = h_1 )
Tpoint_2 = T_{1 sat}
Tpoint_3 = T_{1 sat}
Tpoint_4 = T_{1 sup}
Tpoint_5 = T ( 'R11' , P = P_2 , h = h_3 )
Tpoint_6 = T_2
Tpoint_7 = T_2
Tpoint_8 = T ( 'R11' , P = P_1 , h = h_1 )

Tpoint_{19} = T_{water1}
Tpoint_{20} = T_{water2}

spoint_1 = s ( 'R11' , T = Tpoint_1 , h = h_1 )
spoint_2 = s ( 'R11' , T = Tpoint_2 , x = 0 )
spoint_3 = s ( 'R11' , T = Tpoint_3 , x = 1 )
spoint_4 = s ( 'R11' , T = Tpoint_4 , h = h_2 )
spoint_5 = s ( 'R11' , T = Tpoint_5 , h = h_3 )
spoint_6 = s ( 'R11' , T = Tpoint_6 , x = 1 )
spoint_7 = s ( 'R11' , T = Tpoint_7 , x = 0 )
spoint_8 = s ( 'R11' , P = P_1 , h = h_1 )
A-2 Definition of terms

Here below terms have been used in some parts of this report for economical comparison between different options and technologies.

NPV: Net present value

In order to evaluate the project while considering long term value of money, term “net present value” is considered. It is sum of present values which accounts for incomes minus costs for each period. The more positive the net present value is more value the project would add to the investment.

To calculate “NPV” the term in Equation 6 is used:

\[
\text{Equation 12: } \frac{R}{(1+i)^t}
\]

R stands for net benefit amount in a certain time and “i” is the interest rate. “t” is the time which this term is calculated in. This can be zero before first year, 1 for first year, 2 for second year, etc. NPV can be gained by calculating this term and adding them up for life of project.58

IRR: Internal rate of return

To evaluate profitability of a project” Internal rate of return” is used. This is the interest rate which makes the net present value to be zero. The more “IRR”, the project will return the investment from benefits faster.

To calculate this, the term in Equation 6 is considered for each year, and then the terms are sum up and are equated to zero. “i” is the unknown here, and once it is calculated, this will give the “IRR”.59

PBT: Payback time

Payback time is the duration which the investment returns from benefit of project. It is interested that payback time be as low as possible. It is the ratio between initial investment and annual revenue.


Dry – Isentropic and wet Fluids

Working fluids are classified into three main categories, based on the slope of their saturation curves. They can be regarded to as wet with negative slope, dry with positive slope and isentropic close to infinite slope. In here bellow figures “wet”, “isentropic” and “dry” fluids are shown respectively.\(^6\)

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Figure 30.2 Wet, Isentropic and dry fluid
Source: Wikipedia\(^5^9\)

**Pinch point temperature**

Here by pinch point temperature it is meant: Minimum allowable temperature difference between two streams in different sides of a heat exchanger.

**Gasification**

“Gasification (direct oxidation, starved oxygen combustion) is a process that utilize less than the stoichiometric amounts of oxygen needed for complete combustion. In this process, solid fuels can be converted to a form that can be used more easily. By gasification, biomass is converted to Syngas (CO and H\(_2\)) in gasifier which can then be burned in furnaces, internal combustion engine or gas turbine” (Reza Fakhraie, 2009)\(^6^1\).

**Turbine and Expander**

Both are used to extract power from fluid energy. “Turbines in high inlet-pressure applications are sometimes called expanders. The terms “turbine” and “expander” can be used interchangeably for most applications, but expander is not used when referring to kinetic energy applications, as the fluid does not go through significant expansion” (Daniel Hinch, 2010).\(^6^2\)

For ORC applications three main types of turbines are used: Radial inflow turbines, Euler turbines and axial turbines:

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\(^{61}\) Reza Fakhraie, Renewable energy lecture notes, KTH, 2009.

**Radial inflow turbines:**

In this type of turbine flow enters the turbine in radial direction and after passing the blades will exit in axial direction. They are prone for KW and MW applications of ORC.

![Image of radial inflow turbine](image)

*Figure 31: Blade shape of radial inflow turbine*
*Source: ADT, 2010.*  

**Euler turbines: Radial Outflow turbine**

This is a new type of turbine which is called Euler turbine. As depicted in Figure 36; flow will enter axially and will exit in radial direction. The optimum rotational speed of this type is half of radial inflow turbines. They are robust and erosion resistant (Energent, 2010). They are currently perfect for about less than 500 KWe applications.

![Image of radial outflow turbine](image)

*Figure 32: Radial outflow turbine*
*Source: Energent, 2010.*

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63 ADT, Available at: http://www.adtechnology.co.uk/applications/?page=radial_turbine_impeller; as accessed: 7.10.2010.
Axial turbines:

In this type of turbine, nozzles accelerate the fluid and direct fluid toward rotor. The flow passes in axial direction. They are suitable for higher power ranges in comparison with radial types.

Historically axial turbines where used for geothermal ORC applications. But in case of single stage axial turbine they limit the pressure ratio (Ratio between inlet fluid pressure and outlet fluid pressure). This problem is not encountered in radial inflow turbines. In radial type, working fluid is able to be operated close to its critical pressure (Marcuccilli & Zouaghi, 2007).

Expanders:

Main type of expanders in ORC applications is screw expander. Screw expanders are pair of helical rotors in a casing. During rotation, the trapped volume is increased and this causes expansion. If they rotate in other direction, they work as compressors. If liquid is present, there will not be so much drawbacks on performance and efficiency of the machine. So, they can even admit wet fluid.

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**Turbine losses and mass flow rate:**

Higher mass flow rate in turbine will give better match between fluid velocity at turbine nozzle and rotor tip speed. Moreover, low boiling temperature will give low disc friction losses. These all result in less losses in ORC turbine with comparison to steam turbine. Please refer to S. K. RAY and G. Moss (1965), for more information.

**De-super heater:**

ORC fluid after expansion in turbine or expander is normally in vapor phase. Before it enters the condenser, de-super heater is used to produce saturated vapor working fluid.

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Fuel cell:

It is an electrochemical device which converts chemical energy into electricity (Wikipedia, 2010). Main parts are Anode, cathode and electrolyte. At the anode a catalyst oxidizes the fuel, the ions travel through the electrolyte and at cathode, and the ions are reunited with the electrons. The produced electrons in anodes make an electrical circuit. A basic cell type is shown in Figure 34.

Main types of fuel cells are (Americanhistory, 2008):

**PEM fuel cell**: Proton Exchange Membrane fuel cells have a polymer electrolyte. Efficiency range of this type of fuel cell is 40-50%, operating temperature is about 80°C. Since this type of fuel cell works at low temperatures, they are suitable for homes and vehicles usage. Their fuels need to be purified, and the catalyst which is platinum increases the cost.

**Molten oxide fuel cell**: apply salt components same as sodium or magnesium, and carbonates for its electrolyte. Efficiency is about 60-80%, and working temperature is about 600-700°C. Units with output up to 2 (MW) is built. Increase in temperature is limited since it causes carbon monoxide "poisoning" and this halts the operation.

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67 Americanhistory; available at: [http://americanhistory.si.edu/fuelcells/basics.htm](http://americanhistory.si.edu/fuelcells/basics.htm); as accessed: 19.10.2010.
**Solid oxide fuel cell:** Solid Oxide fuel cells apply ceramic compound of metal same as calcium or oxides as electrolyte. Efficiency is nearly 60 percent, with operating temperatures of 1000°C.

Fuel cells are the cleanest energy conversion systems, however, investment cost of them are fairly high in comparison with conventional gas turbines, internal combustion engines, etc. But, they have high conversion efficiency and this makes them competitive and attractive when long term operation of system is considered. In other word, If system is considered in long run, net present value of typical solid oxide fuel cell and molten oxide fuel cell, is higher than competitive technologies same as gas turbines, Internal combustion engines. Moreover, environmental benefits can be added to this (Roberto Bove, Piero Lunghi, 2006).68

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